CONTENTS

CHAPTER		PAGE
1649	STORY OF CONSOLIDATION	1
1650	**Special Meeting December 1, 1958, Chicago, Ill.**	9
1651	CONSOLIDATION DOCUMENTS	13
1652	**ASHAE Sixty-Fifth Annual Meeting, 1959, Philadelphia, Pa.**	47
1653	RESEARCH ON HEATING AND VENTILATION IN RELATION TO HUMAN COMFORT, BY THOMAS BEDFORD	83
1654	METABOLIC PATTERNS IN STRESSFUL ENVIRONMENTS, BY NELLO PACE	101
1655	ENVIRONMENT REACTIONS IN THE 80 TO 105 F ZONE, BY B. H. JENNINGS AND BARUCH GIVONI	115
1656	SUPPLY OUTLET LOCATIONS FOR BASEMENT HEATING, BY J. R. WRIGHT AND D. R. BAHNFLETH	137
1657	PULSATIONS IN SINGLE-PORT GAS-FIRED RESIDENTIAL HEATING EQUIPMENT, BY C. F. SPEICH AND A. A. PUTNAM	153
1658	LOW FREQUENCY COMBUSTION NOISE IN OIL-BURNING EQUIPMENT, BY G. J. SANDERS AND W. E. LAWRIE	181
1659	OMNI-DIRECTIONAL LOW-SPEED HOT-WIRE ANEMOMETER, BY J. F. KEMP	205
1660	FROST FORMATION AND HEAT TRANSFER ON A CYLINDER SURFACE IN HUMID AIR CROSS FLOW, PART I—EXPERIMENTAL STUDY, BY P. M. CHUNG AND A. B. ALGREN	213
1661	FROST FORMATION AND HEAT TRANSFER ON A CYLINDER SURFACE IN HUMID AIR CROSS FLOW, PART II—THEORETICAL STUDY AND CONCLUSIONS, BY P. M. CHUNG AND A. B. ALGREN	229
1662	DOWNWARD HEAT TRANSFER THROUGH A JOIST SPACE, BY P. M. CHUNG AND C. E. LUND	245
1663	ANNUAL ENERGY USE FOR ELECTRIC SPACE HEATING, BY R. L. BOYD	259
1664	HEATING BY HYPOCAUST, BY EDWIN D. THATCHER	269
1665	PROBE FOR THERMAL CONDUCTIVITY MEASUREMENT OF DRY AND MOIST MATERIALS, BY W. WOODSIDE	291
1666	HEAT GAIN THROUGH WINDOWS SHADED BY METAL AWNINGS, BY NECATI OZISIK AND L. F. SCHUTRUM	311

CHAPTER		PAGE
1667	A New Method for Calculating Radiant Exchanges, by Benjamin Gebhart	321
1668	The Morning After for Education, by Dr. Eric A. Walker	333
1669	**ASHRAE Annual Meeting, 1959, Lake Placid, N. Y.**	341
1670	An Analysis of Turbine Driven Centrifugal Refrigeration Systems, by W. G. Dorsey, Jr.	349
1671	Effects of Refrigerant Properties on Centrifugal Compressor Impeller Dimensions and Stage Performance, by F. J. Wiesner, Jr. and H. E. Caswell	355
1672	Free Piston Compressor in an Air-Conditioning System, by R. J. McCrory, R. W. King, and J. H. McNinch	377
1673	Automatic Computer for Fan Testing, by C. H. Pountney, Jr. and D. W. Skipworth	393
1674	Evaluation of Filters for Removing Irritants from Polluted Air, by N. A. Richardson and W. C. Middleton	401
1675	Fundamental Analysis of Chimney Performance, by W. G. Colborne and W. C. Moffatt	417
1676	A New Method of Chimney Design and Performance Evaluation, by W. C. Moffatt and W. G. Colborne	431
1677	Investigation and Control of Refrigerator Noise, by E. A. Baillif and J. P. Laughlin	443
1678	Some Methods for Investigating Noise from Compressors Used on Household Refrigerators, by R. C. Binder	471
1679	Acoustical Testing of Mufflers for Halogenated Hydrocarbon and Oil Refrigeration Systems, by A. F. Martz, Jr., J. L. Martin, and W. R. Danielson	479
1680	Suppression of Oscillations in Gas-Fired Residential Heating Equipment, by C. F. Speich and A. A. Putnam	489
1681	Solar Radiant Gains Through Directional Glass Exposure, by R. D. Cramer and L. W. Neubauer	499
1682	Load Calculations Using Pretabulated Admittance Functions, by W. B. Drake, Harry Buchberg, and D. Lebell	515
1683	Transfer Admittance Functions for Typical Composite Wall Sections, by W. B. Drake, Harry Buchberg, and D. Lebell	523

CHAPTER		PAGE
1684	LARGE-SCALE WALL HEAT-FLOW MEASURING APPARATUS, BY K. R. SOLVASON	541
1685	CONDENSATION BETWEEN THE PANES OF DOUBLE WINDOWS, BY A. GRANT WILSON AND E. S. NOWAK	551
1686	EVAPORATIVE COOLING FOR COMMON STORAGES OF FRUITS AND VEGETABLES, BY R. S. ASH	571
1687	CONTROLLED ATMOSPHERE APPLE STORAGE PROCESS AND ITS REQUIREMENTS ON REFRIGERATION STRUCTURES AND SYSTEMS, BY J. W. ZAHRADNIK	589
1688	PROPOSED ASHRAE STANDARDS FOR THE MEASUREMENT OF SOUND FROM EQUIPMENT, BY C. M. ASHLEY	601
1689	EXPERIMENTAL STUDY OF GRILLE NOISE CHARACTERISTICS, BY B. H. MARVET	613
1690	EFFECT OF FUEL COMPOSITION ON OIL BURNER NOISE, BY W. A. BEACH, R. W. SAGE, AND H. F. SCHROEDER	627
1691	COMPARISON OF FLUOROALKANE ABSORPTION REFRIGERANTS, BY B. J. EISEMAN, JR.	635
1692	ESTIMATING WATER CONTENT OF CERTAIN DRIED BUT UNCHARGED HERMETIC REFRIGERATING COMPRESSORS, BY H. M. ELSEY, J. B. KELLEY, AND R. B. SHARPE	651
1693	EFFECT OF PRESSURE AND TEMPERATURE ON THE VISCOSITY OF REFRIGERANTS IN THE VAPOR PHASE, BY O. W. WITZELL AND C. Z. KAMIEN	663
1694	REPEATED SCRAPE ABRASION TESTING OF ENAMELED WIRES IN GASEOUS AND LIQUID REFRIGERANTS, BY F. F. TRUNZO, G. J. BICH, AND G. W. HEWITT	675
1695	DETERMINATION OF THE EFFECTIVENESS OF WINDOW SHADING MATERIALS ON THE REDUCTION OF SOLAR RADIATION HEAT GAIN, BY R. C. JORDAN AND J. L. THRELKELD	683
1696	HEAT FLOW THROUGH GLASS WITH ROLLER SHADES, BY NECATI OZISIK AND L. F. SCHUTRUM	697
1697	EVALUATION OF THREE ROOM AIR DISTRIBUTION SYSTEMS FOR SUMMER COOLING, BY J. J. REINMANN, ALFRED KOESTEL, AND G. L. TUVE	717
1698	SURFACE ODOR ADSORPTION AND RETENTION PROPERTIES OF SURFACES, BY W. F. HOPPER	735
	In Memoriam	745
	WALTER L. FLEISHER (1881–1959)	747
	GEORGE A. HORNE (1877–1959)	749
	Index	751

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Regions Central: Walter A. Grant, *Chairman;* R. H. Tull, *Vice Chairman;* T. J. Phillips (I); C. C. E. Harris (I); R. A. Baker (I); W. G. Hole (II); J. W. Chandler (III); John Everetts, Jr. (III); G. B. Rottman (III); J. G. Woodroof (IV); T. G. Crider (V); W. E. Fontaine (V); P. J. Marschall (VI); J. B. McShane (VI); G. H. Poggen, Jr. (VI); J. W. May (VII); G. W. F. Myers (VII); W. F. Wischmeyer (VII); W. J. Collins, Jr. (VIII); P. N. Vinther (VIII); Fred Janssen (IX); C. L. Hall (X); W. O. Stewart (X).

Advisory Board: Cecil Boling, *Chairman;* C. M. Ashley, Lester T. Avery, J. G. Bergdoll, Jr., M. F. Blankin, Leon Buehler, Jr., P. B. Christensen, S. E. Dibble, S. H. Downs, E. O. Eastwood, H. D. Edwards, Crosby Field, H. P. Gant, P. B. Gordon, Van R. H. Greene, John E. Haines, W. R. Hainsworth, Harry Harrison, H. M. Hart, C. V. Haynes, A. J. Hess, N. H. Hiller, C. F. Holske, G. E. Hulse, L. N. Hunter, John W. James, B. H. Jennings, R. C. Jordan, C. S. Leopold, L. L. Lewis, S. R. Lewis, C. R. Logan, R. H. Money, Glen Muffly, A. W. Oakley, Alfred J. Offner, Fred Ophuls, E. R. Queer, F. B. Rowley, L. E. Seeley, Edward Simons, Harry Sloan, H. F. Spoehrer, A. E. Stacey, Jr., A. B. Stickney, J. F. Stone, Reg. F. Taylor, Henry Torrance, Jr., G. L. Tuve, A. C. Willard.

Chapters Regional Committees* (June 1959–February 1960)

Regional Areas	*Chapter*	*Delegate*	*Alternate*
Region I—T. J. PHILLIPS, *Coordinator*			
N. Y.	Long Island (Garden City)	C. M. Alston	S. M. Walzer
N. Y.	New York	L. D. Carr	J. M. Morse
N. J., N. Y.	North Jersey (Newark)	L. G. Huggins	G. A. Freeman
N. Y., Vt.	Northeastern New York (Albany)	M. E. Waddell	R. B. Taylor
C. E. HARRIS, *Regional Director*			
Mass., N. H., Me.	Boston	F. J. Butler	W. S. Warner
N. Y.	Central New York (Syracuse)	L. D. Carr	J. M. Morse
N. Y.	Niagara Frontier (Buffalo)	C. W. Stone	D. J. Seifert
Conn.	Northern Connecticut (Hartford)	Walter Heywood	F. J. Raible, Jr.
R. I., Mass.	Rhode Island (Providence)	J. P. Delaney	C. H. Dow
N. Y.	Rochester	L. C. Engelhart	H. J. Dyminski
Conn., N. Y.	Southern Connecticut (New Haven)	C. W. McElroy	R. B. Cahoon
Mass.	Western Massachusetts (Springfield)	J. J. Curran	Charles Martin, Jr.

† Resigned as of November 5, 1959.
* The committees listed here and on p. xxi held their regional meetings during the fall of 1959.

Regional Areas	*Chapter*	*Delegate*	*Alternate*
Region III—JOHN EVERETTS, JR., *Coordinator*			
M ., Del., Pa.	Baltimore	W. H. Kruger	W. S. Rock
Pa.	Central Pennsylvania (Harrisburg)	L. R. St. Onge	C. W. Prey
Md., Pa.	Johnstown	Sigmond Moroh	J. K. Thornton
Md., Va., W. Va., D. C.	National Capital (Washington, D. C.)	J. J. Nolan, Jr.	H. E. Grossman
Del., N. J., Pa.	Philadelphia	E. K. Wagner	W. F. Spiegel
Pa., W. Va.	Pittsburgh	J. G. Herrmann	C. W. Stanger
Va., W. Va.	Richmond	G. B. Alexander	J. C. Hildebrand, Jr.
Va.	Hampton Roads (Norfolk)	J. R. Spencer	J. Hart
Region IV—J. G. WOODRUFF, *Coordinator*			
Ga.	Atlanta	B. W. Dean	J. P. McLaney
Fla.	Florida West Coast (Tampa)	S. F. Graziano	Leland Desmon
Ga., Fla.	Jacksonville	J. M. Hammond	H. L. McMurray
Ga., S. C.	Savannah	R. A. Craig, Jr.	E. F. White
Fla.	South Florida (Miami)	A. R. Martin, Jr.	Hugh Kirkpatrick
G. B. ROTTMAN, *Regional Director*			
N. C.	North Piedmont (Greensboro)	B. Jeglinski	A. J. Lane
S. C.	South Carolina (Columbia)	B. A. Leppard	H. J. Haar, Jr.
N. C.	South Piedmont (Charlotte)	W. P. Wells	J. R. Clark
Region V—T. G. CRIDER, *Coordinator*			
Ohio	Cleveland	J. L. Frissee	R. M. Hepner
Ohio, W. Va.	Columbus	R. C. Liebert	P. G. Facey, Jr.
Ind., Ohio, Mich.	Toledo	J. F. Guest	F. A. Edgington
W. E. FONTAINE, *Regional Director*			
Ind.	Central Indiana (Indianapolis)	J. G. Thornburg	J. K. Teskoski
Ind., Ky., Ohio	Cincinnati	C. W. Couch	George Winkelman
Ind., Ohio	Dayton	T. A. Lutz	F. H. Doench, Jr.
Ill., Ind., Ky.	Evansville	R. E. Deaux	R. E. Ahlf
Region VI—P. J. MARSCHALL, *Coordinator*			
Ill., Ind.	Illinois (Chicago)	J. W. Hall	E. P. Heckel, Jr.
Ill., Iowa	Illinois–Iowa (Moline)	E. O. Hull	R. F. Demange
Iowa	Iowa (Des Moines)	R. S. Stover	C. B. Campbell
Minn., N. D., S. D., Wis.	Minnesota (Minneapolis)	T. D. Merchants	R. G. Gridley
Wis.	Wisconsin (Milwaukee)	R. O. Hardwick	J. E. Illingworth
F. Y. CARTER, *Consultant*			
Mich.	Central Michigan (Battle Creek)	L. R. Smith	F. P. Crotser
Mich.	Michigan (Detroit)	D. S. Falk	M. O. Wehmeyer
Ind., Mich.	Western Michigan (Grand Rapids)	D. A. Rackliffe	Wm. Wessels
Region VII—J. W. MAY, *Coordinator*			
Ky., Ind.	Louisville	H. L. Carr	J. D. Williams, Jr.
La.	New Orleans	T. A. Stokes	E. H. Sanford
La.	Baton Rouge	W. J. Le Blanc	A. J. Mayers, Jr.
W. F. WISCHMEYER, *Regional Director*			
Mo.	Kansas City	L. A. Heaven	A. J. Sneller
Fla., Ala.	Mobile	W. J. Schilling, Jr.	G. M. Elliott
Ala.	North Alabama (Birmingham)	J. H. Judd	G. R. Jackson
G. W. F. MYERS, *Regional Director*			
Tenn., Ark., Miss., Mo.	Memphis	H. W. Wade	C. L. Brown, Jr.
Tenn.	Middle Tennessee (Nashville)	J. R. Potter	D. E. Nichols
Miss.	Mississippi (Jackson)	Burt Lomax, Jr.	D. F. Ingram
Ill., Mo.	St. Louis	K. O. Williams	F. G. Meyers

Long Range Planning: John E. Haines, *Chairman;* C. M. Ashley, Leon Buehler, Jr., John W. James, J. Donald Kroeker, H. F. Spoehrer.

Admissions and Advancement: T. J. Phillips, *Chairman;* P. R. Achenbach, *Vice Chairman;* F. H. Buzzard, G. J. Finck, H. D. Glaser, C. F. Kayan, C. S. Koehler.

Charter and By-Laws: F. H. Faust, *Chairman;* J. C. Fitts, *Vice Chairman;* Walter Cooke.

Education: W. L. Rose, *Chairman;* Milton Kalischer, *Vice Chairman;* J. B. Chaddock, J. H. Spence.

Exposition: L. N. Hunter, *Chairman;* W. H. Aubrey, *Vice Chairman;* S. H. Downs, J. E. Dube, John W. James, F. L. Tarleton, P. K. Barker, *ex officio.*

Honors and Awards: W. L. Holladay, *Chairman;* Leon Buehler, Jr., C. Rollins Gardner, C. C. Grote, J. G. Woodroof.

International Relations: W. T. Pentzer, *Chairman;* C. M. Ashley, P. B. Christenson, J.E. Dube, R. C. Jordan, C. F. Kayan, Alfred J. Offner, C. H. Pesterfield.

Meetings Arrangements: P. K. Barker, *Chairman;* C. B. Gamble, P. J. Marschall, L. N. Hunter, *ex officio.*

Membership Development: K. M. Newcum, *Chairman;* G. W. F. Myers, J. H. Ross, H. P. Tinning, F. L. Vaughn, V. D. Wissmiller.

Nominating Committee: P. B. Gordon, *Chairman;* Leon Buehler, Jr.*

Selected by Chapters Regional Committees:

Region	*Member*	*Alternate*
I	S. F. Gilman	John Bonner
II	G. C. Davis	J. C. Lowcock
III	W. A. Siegfried	G. B. Alexander
IV	R. K. Rouse	W. B. Knox
V	L. C. Burkes	R. S. Buchanan
VI	D. S. Falk	J. H. Jester
VII	D. M. Mills	Walter Cooke
VIII	G. H. Meffert	F. R. Denham
IX	J. K. James	J. V. Berger
X	R. M. Westcott	W. B. Morrison

Selected by Board of Governors:

Area of Membership Interest	*Member*	*Alternate*
1. Heating	R. A. Sherman (V) B. W. Farnes (X)	A. F. Hubbard (VI) M. W. McRae (VI)
2. Refrigeration	R. A. Baker (I) W. L. McGrath (I)	P. B. Christensen (I) J. R. Caulk, Jr. (VII)
3. Air Conditioning & Ventilation	H. G. S. Murray (II) P. N. Vinther (VIII)	C. R. Gardner (VIII) Walter Cooke (VII)

Professional Development: C. M. Ashley, *Chairman;* Leon Buehler, Jr., *Vice Chairman;* Crosby Field, H. P. Hayes, R. S. Stover.

Program: S. J. Williams, Jr., *Chairman;* P. N. Vinther, *Vice Chairman;* H. S. Buchanan, W. E. Fontaine, H. P. Harle, W. G. Hole, R. A. Line, J. W. May, W. L. McGrath, Justin Neuhoff, G. B. Priester.

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* Member from last prior Nominating Committee.

Publications: John Everetts, Jr., *Chairman;* J. W. May, *Vice Chairman;* W. P. Chapman, P. B. Christensen, H. G. Gragg, H. M. Hendrickson,† J. P. McDermott,‡ G. R. Munger, C. W. Phillips, G. B. Priester, W. L. Ross, S. C. Segal, R. H. Tull, L. K. Warrick.

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JOURNAL Subcommittee: J. W. May, *Chairman:* W. J. Collins, Jr.

GUIDE and DATA BOOK Subcommittee: P. B. Christensen, *Chairman.*

Applications: P. B. Christensen, *Chairman;* F. H. Buzzard, W. G. Hole, S. P. Soling.
Equipment: W. L. McGrath, *Chairman;* J. L. Ditzler, P. J. Marschall, V. D. Wissmiller.
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Consultants to Committee: P. R. Achenbach, A. P. Boehmer, D. L. Fiske, Albert Giannini, W. C. L. Hemeon, E. N. Johnson, A. M. G. Moody, Maurice Nelles, H. B. Nottage, F. G. Peck, E. J. Robertson, P. H. Yeomans.

Research Advisory Committees: (for complete listing see p. xvi).

Standards: P. W. Wyckoff, *Chairman;* P. N. Vinther, *Vice Chairman;* J. R. Caulk, Jr., K. T. Davis, A. S. Decker, J. L. Ditzler, Albert Giannini, Fred Janssen, J. D. Loveley, W. L. McGrath, G. B. Priester, F. J. Reed, J. A. Schenk, S. P. Soling, Herbert Wolf.

METHOD OF TESTING FOR RATING ROOM AIR CONDITIONERS* (Standard No. 16-56R): R. H. Meyerhans, *Chairman;* C. G. Coyne, K. D. Crook, R. W. Gilmer, E. Gmoser, P. E. Kolb, J. R. Schreiner, E. H. Schwenker, E. C. Tanner. *Liaison:* W. L. McGrath.

METHODS OF TESTING FOR RATING THERMOSTATIC AND CONSTANT PRESSURE REFRIGERANT EXPANSION VALVES* (ASA B60.1-1950; Standard No. 17-48R): D. C. Albright, *Chairman;* F. Y. Carter, K. E. Wilson, W. F. Wiselmeyer. *Alternates:* Willis Stafford, D. S. Sterner. *Liaison:* J. A. Schenk.

METHOD OF TESTING FOR RATING REMOTE AIR-COOLED EVAPORATIVE CONDENSERS* (Standard No. 20-41R): R. H. Merrick, *Chairman;* R. J. Campbell, C. Collins, C. A. Dubberly, T. F. Facius, H. E. Kunkel, R. C. Lower, O. J. Nussbaum, M. A. Ramsey, L. R. Smith, G. Wood, *Liaison:* A. S. Decker.

METHOD OF TESTING FOR RATING WATER-COOLED REFRIGERANT CONDENSERS* (Standard No. 22-48R): L. R. Smith, *Chairman;* P. K. Barker, J. G. Bergdoll, C. E. Drake, R. L. Jones, W. O. Kline. *Liaison:* A. S. Decker.

METHODS OF TESTING FOR RATING LIQUID COOLERS* (Standard No. 24-57R): R. L. Jones, *Chairman;* R. A. Mayer, *Secretary;* P. K. Barker, J. D. Bergdoll, L. R. Smith. *Liaison:* A. S. Decker.

† Resigned as of November 5, 1959.
‡ Filled unexpired term.
* Project Committee for Proposed Revisions.

GENERAL AND SPECIAL COMMITTEES—June 1959–February 1960

METHOD OF TESTING FOR RATING RETURN-LINE LOW-VACUUM HEATING PUMPS* (Standard No. 47-35R): W. M. Wallace, II, *Chairman*; H. E. Adams, H. S. Bean, A. H. Church, E. G. Keller, L. M. Nechine, A. O. Roche, Jr., L. E. Stearns.

METHOD OF TESTING FOR RATING STEAM UNIT VENTILATORS* (Standard No. 53-34R): J. W. McElgin, *Chairman*; L. A. Cherry, A. F. Hubbard, D. H. Krans, L. G. Miller.

METHOD OF TESTING FOR RATING LIQUID CHILLING PACKAGES† (Standard No. 30P): L. R. Smith, *Chairman*; R. L. Jones, *Secretary*; P. K. Barker, C. S. Cave, Arthur Courtney, C. E. Drake, W. O. Kline, D. H. Krans, R. A. Mayer, S. B. Nissley, D. D. Stover. *Liaison:* A. S. Decker.

METHOD OF TESTING FOR RATING SOLENOID VALVES FOR REFRIGERATION AND AIR-CONDITIONING SYSTEMS† (Standard No. 31P): J. A. Schenk, *Chairman*; D. C. Albright, F. Y. Carter, C. C. Hansen, III, D. S. Sterner, W. F. Wischmeyer. *Alternates:* J. M. Strauss, Keith Wilson. *Liaison:* J. R. Caulk, Jr.

METHODS OF EQUIPMENT SOUND TESTING† (Standard No. 36P): C. M. Ashley, *Chairman*; W. E. Blazier, J. B. Chaddock, A. F. Hubbard, R. D. Madison, S. P. Soling. *Consultant:* R. J. Wells. *Secretary:* W. R. Kerka. *Liaison:* S. P. Soling.

METHODS OF TESTING FOR RATING UNITARY AIR-CONDITIONING EQUIPMENT† (Standard No. 37P): J. R. Schreiner, *Chairman*; H. T. Gilkey, G. Keane (AMCA), R. C. Lower, W. H. Mullin, W. A. Sopfford, W. E. Ural. *Liaison:* W. L. McGrath.

METHODS OF TESTING FOR RATING UNITARY HEAT PUMPS FOR AIR CONDITIONING† (Standard No. 39P): G. L. Biehn, *Chairman*; R. N. Mahendra, *Secretary*; C. H. Belt, H. W. Jobes, R. C. Lower, R. G. McReady, W. A. Spofford. *Liaison:* W. L. McGrath.

METHODS OF TESTING FOR RATING HEAT-OPERATED UNITARY AIR CONDITIONERS† (Standard No. 40P): J. G. Reid, Jr., *Chairman*; D. E. Heath, R. H. Merrick, B. A. Phillips, F. J. Reed. *Liaison:* W. L. McGrath.

STANDARD MEASUREMENTS† (Standard No. 41P): C. W. Phillips, *Chairman*; F. J. Reed, W. A. Spofford, G. Thompson. *Liaison:* F. J. Reed.

STANDARD FOR COMFORT AIR CONDITIONING† (Standard No. 42P): M. K. Fahnestock, *Chairman*; *Liaison:* S. P. Soling.

MINIMUM OUTDOOR AIR REQUIREMENTS FOR AIR CONDITIONING† (Standard No. 49P): Albert Giannini, *Chairman*; C. B. Gamble, R. A. Gonzalez, R. J. Salinger, H. G. Strong.

METHOD OF TESTING FOR RATING EVAPORATIVE COOLERS† (Standard No. 50P): R. S. Ash, *Chairman*; S. F. Duncan, H. S. Dutcher, E. V. Gritton, Leo Hungerford, C. B. Rowe, W. T. Smith, J. R. Watt.

STANDARD FOR EVALUATION OF AIR CLEANING DEVICES FOR GENERAL VENTILATION AND AIR CONDITIONING† (Standard No. 52P): P. R. Achenbach, *Chairman*; W. C. L. Hemeon, R. F. Logsdon, P. E. McNall, Jr.

METHOD OF TESTING FOR RATING GRAVITY ROOF VENTILATORS† (Standard No. 54P): R. G. Nevins, *Chairman*; G. C. Breidert, F. N. Calhoon, R. B. Engdahl, W. V. Hukill, J. P. Johnson, K. E. Robinson, F. B. Rowley, L. J. Shumaker.

SPECIAL COMMITTEES—June 1959–February 1960

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Building: P. J. Marschall, *Chairman;* C. M. Ashley, M. F. Blankin, R. C. Jordan, R. A. Sherman, J. F. Stone.

Emblem and Insignia: D. H. McCuaig, *Chairman;* P. J. Marschall, M. C. Turpin.

ASHRAE-AIA Cooperation: John E. Haines, *Chairman;* C. B. Gamble, A. J. Hess, H. C. Hoffmann, G. A. Linskie, R. J. Salinger, J. L. Wolf, P. H. Yeomans.

Steering: *Staff and Administration*—Cecil Boling, P. B. Gordon, E. R. Queer.
Finance—Cecil Boling, F. Y. Carter, J. H. Fox, E. R. Queer.
Non-Technical and Regional—J. H. Fox, Walter A. Grant, D. D. Wile.
Technical—John Everetts, Jr., H. F. Spoehrer, R. H. Tull.

UEC Fund Raising: M. F. Blankin, *Chairman;* John E. Haines, *Vice Chairman;* T. E.Brewer, C. F. Holske.

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* Project Committee for Proposed Revisions.
† Project Committee for Proposed Standards.

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Vice President.....T. H. Collard, Jr.
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Secretary.....R. W. Hole
Treasurer.....William Baker
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Vice President.....J. R. Hight
Secretary.....S. A. Frederickson
Treasurer.....W. C. Miessemer
Board of Governors: S. A. Frederickson, F. W. Gabbard, J. R. Hight, G. L. Jackson, Jr., W. C. Miessemer

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Co-President.....John Thornburgh
Vice President.....A. B. Keller
Co-Vice President.....W. F. Freije, Jr.
Secretary.....D. J. Thomas
Co-Secretary.....W. L. Kercheval
Treasurer.....E. E. Hunter
Board of Governors: W. F. Currise, W. F. Freije, Jr., E. E. Hunter, A. B. Keller, Lee Kennedy, W. L. Kercheval, J. M. Teskoski, D. J. Thomas, John Thornburgh

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Secretary.....W. E. Macbeth
Treasurer.....R. E. Price
Board of Governors: F. P. Crotser, W. M. Hassenplug, W. E. Macbeth, E. D. Milway, R. E. Price, L. R. Smith, R. B. Smith

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* Served first half of year.
† Served last half of year.

Local Chapter Officers—1959 (*Continued*)

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2nd Vice President . . . . . L. L. Schneider
3rd Vice President . . . . . R. A. Barr
4th Vice President . . . . . E. L. Galson
Secretary . . . . . B. P. Morabito
Treasurer . . . . . D. C. Raasch
Board of Governors: R. A. Barr, E. L. Galson, K. M. Gerteis, S. F. Gilman, W. A. La Grange, B. P. Morabito, E. L. Moyer, D. C. Raasch, L. L. Schneider

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Headquarters, Oklahoma City

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Vice President . . . . . S. R. Hill, R. R. Ellis*
Secretary . . . . . J. T. McKinney
Treasurer . . . . . R. G. Keif
Board of Governors: F. R. Denham, W. R. Johnson, A. C. Shelley

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Treasurer . . . . . R. C. Niess
Assistant Treasurer . . . . . A. E. Diehl
Board of Governors: A. E. Diehl, H. L. Hull, R. C. Niess, R. F. Reagan, E. P. Short, Jr.

Chapitre de la Ville de Quebec

Headquarters, Quebec, P.Q., Canada

President . . . . . P. J. Lamarche
Vice President . . . . . Francois L'Anglais
Secretary . . . . . Az. Servant
Treasurer . . . . . Jean-Paul Boulay
Board of Governors: Louis-Philippe Bonneau, Jean-Marc Lagace, Lucien Larocque, Richard Maillette, Maurice Paquet

Cincinnati

Headquarters, Cincinnati, Ohio

President . . . . . C. W. Couch
Vice President . . . . . George Winkelman
Secretary . . . . . C. P. Wood, Jr.
Treasurer . . . . . C. J. Kummer
Board of Governors: C. W. Couch, C. P. Krantz, C. J. Kummer, N. E. Rau, W. H. Rieger, F. N. Thompson, George Winkelman, C. P. Wood, Jr.

Cleveland

Headquarters, Cleveland, Ohio

President . . . . . R. D. Wilson
Vice President . . . . . R. M. Hepner
Secretary . . . . . R. M. Rubin
Treasurer . . . . . D. J. Noeth
Board of Governors: P. E. Davis, B. A. Schwirtz, R. A. Urban

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* Filled unexpired term.

Columbus

Headquarters, Columbus, Ohio

President . . . . . J. W. Ford
1st Vice President . . . . . Austin Jones
2nd Vice President . . . . . E. T. Stluka
Secretary . . . . . G. H. Reverman
Assistant Secretary . . . . . R. L. Keener
Treasurer . . . . . J. W. Hensel
Board of Governors: P. G. Facey, R. C. Liebert, N. K. Mason

Dallas

Headquarters, Dallas, Tex.

President . . . . . J. J. Mays, Jr.
Vice President . . . . . O. H. Mehl
Secretary . . . . . R. M. Kilpatrick
Treasurer . . . . . F. L. McFadden, Jr.
Board of Governors: J. D. Brock, W. P. Dickson, W. P. Glancy, J. D. Poythress, Rose Zumwalt

Dayton

Headquarters, Dayton, Ohio

President . . . . . N. O. Mitchell
Vice President . . . . . T. A. Lutz
Secretary . . . . . F. H. Doench, Jr.
Treasurer . . . . . D. G. Ely
Board of Governors: L. P. Brehm, R. L. Brugler, J. L. Homan, R. B. Walcott

El Paso

Headquarters, El Paso, Tex.

President . . . . . J. A. True
Vice President . . . . . J. H. Brooks
Secretary . . . . . J. L. Huff
Treasurer . . . . . Roy Gibson
Board of Governors: M. D. Goodwin, F. A. Huchet, Jr., G. H. Jackson, J. W. Lantow

Evansville

Headquarters, Evansville, Ind.

President . . . . . R. E. Deaux
1st Vice President . . . . . R. E. Ahlf
2nd Vice President . . . . . H. C. Shagaloff
3rd Vice President . . . . . C. L. Herndon
Secretary . . . . . D. S. Phillips
Treasurer . . . . . K. C. Davis
Board of Governors: R. P. Garvey

Florida West Coast

Headquarters, Tampa

President . . . . . J. C. Johnson, Jr.
Vice President . . . . . K. W. Whittington
Secretary . . . . . Roy Hern
Treasurer . . . . . C. L. Dugan, D. C. McNeil*
Board of Governors: W. A. Pitt, Jr., Richard Stenholm, F. C. Whitaker

Fort Worth

Headquarters, Fort Worth, Tex.

President . . . . . J. K. Mattox, Jr.
Vice President . . . . . C. L. Zahn
Secretary . . . . . G. F. Smith
Treasurer . . . . . J. R. Blanke
Board of Governors: A. L. Freer, P. P. Schwartz

Local Chapter Officers—1959 (*Continued*)

Golden Gate

Headquarters, San Francisco, Calif.

President.....T. R. Simonson
Vice President.....Karl Guttmann
Secretary.....R. C. Pribuss
Treasurer.....D. A. Delaney
Board of Governors: L. E. Dwyer, J. D. Kniveton

Hampton Roads

Headquarters, Norfolk, Va.

President.....J. R. Spencer
Vice President.....R. F. Fox
Secretary.....W. B. Britton
Treasurer.....J. A. Hoffman
Board of Governors: D. L. Gusler

Houston

Headquarters, Houston, Tex.

President.....H. D. McMillan, Jr.
1st Vice President.....J. R. Jones
2nd Vice President.....E. H. McLane
Secretary.....J. B. Buckley
Treasurer.....T. W. Leatherwood
Board of Governors: H. B. Edwards, K. C. Gruber, F. R. Young

Illinois

Headquarters, Chicago

President.....J. W. Hall
Vice President.....E. P. Heckel, Jr.
Secretary.....W. V. Richards
Treasurer.....L. K. Warrick
Board of Governors: E. N. Johnson, Herbert Kreisman, D. A. Parkhurst

Illinois-Iowa

Headquarters, Moline, Ill.

President.....E. O. Hull
Vice President.....R. F. Demange
Secretary.....D. G. Johnson
Treasurer.....M. L. Smith
Board of Governors: W. M. Curtin, P. J. Hannon

Inland Empire

Headquarters, Spokane, Wash.

President.....J. R. Morris
Vice President.....R. B. Campbell
Secretary.....J. L. Harvey
Treasurer.....D. E. Arnold
Board of Governors: J. S. Latenser, F. J. Lund, R. D. Nevers

Iowa

Headquarters, Des Moines

President.....C. B. Campbell
Vice President.....J. R. Bain
Secretary.....V. G. Polly
Treasurer.....E. C. Giberson
Board of Governors: L. E. Gnade, R. T. Howard, Jr., R. H. Sears, M. B. Townsend

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* Filled unexpired term.

Jacksonville

Headquarters, Jacksonville, Fla.

President.....J. M. Hammond
Vice President.....L. H. Boissoneault
Secretary.....P. E. Stewart
Treasurer.....A. F. Duffek
Board of Governors: F. J. Beckwith, H. L. McMurray

Johnstown

Headquarters, Johnstown, Pa.

President.....Sigmond Moroh
Vice President.....J. K. Thornton
Secretary.....E. S. Spangler
Treasurer.....G. C. MacAlarney
Board of Governors: H. F. Lenz, K. O. Schlentner, L. F. Tierney

Kansas City

Headquarters, Kansas City, Mo.

President.....L. A. Heaven
1st Vice President.....A. J. Sneller
2nd Vice President.....U. M. Scurlock
Secretary.....E. M. Hopkins
Treasurer.....Ernest Engelhardt
Board of Governors: W. S. Clark, J. S. Dukelow, J. C. Fasnacht, P. C. Leffel, Paul Shapiro

Long Island

Headquarters, Garden City, L. I., N. Y.

President.....C. M. Alston
Vice President.....S. M. Walzer
Recording Secretary.....S. L. Gayle
Financial Secretary.....W. T. Cleland
Treasurer.....W. H. Hoops
Board of Governors: H. J. Campbell, Jr., K. F. Henry, W. G. Kane

Louisville

Headquarters, Louisville, Ky.

President.....R. V. Prucha
1st Vice President.....R. F. Logsdon
2nd Vice President.....R. W. Rademacher
Secretary.....H. B. Abbott
Treasurer.....K. J. Roy
Board of Governors: R. W. Anderson, H. L. Carr, D. J. Massa, J. O. Sutter, J. D. Williams

Manitoba

Headquarters, Winnipeg, Man., Canada

President.....W. L. Algie, N. B. Jorgensen*
Vice President.....H. R. Skinner
Secretary.....A. Leiterman
Treasurer.....F. J. Lindenschmidt
Board of Governors: J. L. Greer, Stanley Hayden, D. S. Swain

Memphis

Headquarters, Memphis, Tenn.

President.....H. W. Wade
Vice President.....C. L. Brown, Jr.
Secretary.....Phillip Hall
Treasurer.....A. L. Barksdale
Board of Governors: R. H. Bolding, C. L. Brown, Jr., G. C. Crow, Jr., W. H. Egbert, Jr., D. M. Mills, H. H. Wilson

Local Chapter Officers—1959 (*Continued*)

Michigan

Headquarters, Detroit

President.....R. E. Lee,* J. B. Olivieri†
Vice President..C. J. Henstock,* M. O. Wehmeyer†
Secretary.....J. G. Black, Jr.
Treasurer.....R. E. Maund
Board of Governors: F. J. Drogosch, J. B. Koch, K. A. Nesbitt, K. J. Wagoner

Middle Tennessee

Headquarters, Nashville

President.....I. C. Thomasson
Vice President.....R. L. King
Secretary.....J. R. Potter
Treasurer.....E. W. Moats
Board of Governors: R. L. Bibb, Jr., Faulkner Hickerson, Edward Kennedy

Minnesota

Headquarters, Minneapolis

President.....R. G. Gridley
1st Vice President.....D. S. Swanson
2nd Vice President.....J. W. McNamara
Secretary.....L. D. Freedland
Treasurer.....A. E. Swanberg
Board of Governors: L. D. Freedland, R. G. Gridley, D. A. Hargrave, J. H. Jester, J. W. McNamara, T. D. Merchant, A. E. Swanberg, D. S. Swanson

Mississippi

Headquarters, Jackson

President.....F. H. North
Vice President.....W. D. Fortner
Secretary.....C. E. Strahan, Jr.
Treasurer.....W. F. King
Board of Governors: C. M. Broad, D. F. Ingram, B. M. Meyers, Jr.

Mobile

Headquarters, Mobile, Ala.

President.....W. J. Schilling
1st Vice President.....Jack Chichester
2nd Vice President.....G. M. Elliott
Secretary.....O. S. Posey
Treasurer.....K. E. Buck
Board of Governors: K. E. Buck, Jack Chichester, G. M. Elliott, O. S. Posey, W. J. Schilling, J. P. Scott

Montreal

Headquarters, Montreal, Que., Canada

President.....H. G. S. Murray
1st Vice President.....G. E. Forget
2nd Vice President.....W. E. Jarvis
3rd Vice President.....André Poulin
Secretary.....W. U. Robinson
Treasurer.....H. B. Cooper
Board of Governors: Ian Campbell, Gordon Cox, Arthur de Breyne, J. G. Le Francois, J. C. E. Lowcock, D. C. McKeen

* Served first half of year.
† Served last half of year.

National Capital

Headquarters, Washington, D. C.

President.....J. J. Nolan, Jr.
Vice President.....H. E. Grossman
Secretary.....L. D. Pain
Assistant Secretary.....R. J. Ruschell
Treasurer.....W. C. Hansen
Board of Governors: A. H. Mister, J. W. Morgan, H. W. Rush, H. W. Sweeny

Nebraska

Headquarters, Omaha

President.....F. P. Manchester, Jr.
Vice President.....W. L. Ryan
Secretary.....C. L. Thomsen
Treasurer.....R. W. Scott
Board of Governors: V. L. Crane, Donald Fry, T. W. Peterson

New Mexico

Headquarters, Albuquerque

President.....E. L. Brazier, Jr.
Vice President.....R. P. Lee
Secretary.....V. J. Stephens
Treasurer.....H. V. Frankenfeld
Board of Governors: J. L. Desilets, W. L. Jones, L. E. Meyer

New Orleans

Headquarters, New Orleans, La.

President.....T. A. Stokes
Vice President.....E. H. Sanford
Secretary.....R. D. Lewis
Treasurer.....J. H. Maloney
Board of Governors: E. E. Spranley, E. K. Strahan, Jr., W. W. Taylor, Jr.

New York

Headquarters, New York

President.....L. D. Carr
1st Vice President.....J. M. Morse
2nd Vice President.....S. A. Spencer
Recording Secretary.....H. F. Burpee
Financial Secretary.....P. A. Bourquin
Treasurer.....P. R. Lynn
Board of Governors: G. J. Finck, W. T. Kane, Leonard Kowadlo, Sherman Loud, W. J. Olvany, J. M. Pennesi, H. P. Tinning

Niagara Frontier

Headquarters, Buffalo, N. Y.

President.....D. J. Seifert
Executive Vice President.....A. C. Little
1st Vice President.....F. R. Collins, Jr.
2nd Vice President.....R. W. Bartsch
Secretary.....H. J. McLaughlin
Treasurer.....A. F. Worden, Jr.
Board of Governors: C. R. Fagerstrom, J. P. Guerra, R. H. Mollenberg, C. W. Stone, Q. P. Thompson, J. S. Witter

Local Chapter Officers—1959 (*Continued*)

North Alabama

Headquarters, Birmingham

President.....F. L. Hardy
1st Vice President.....G. S. Lawrence
2nd Vice President.....G. R. Jackson
Secretary.....J. H. Judd
Treasurer.....C. J. Balli
Board of Governors: P. H. Brown, Drexel Daily, L. H. Eberdt, Jr., Jerry Ezkew, G. W. Hogan, Jr., S. S. Simpson, W. L. Wayman

North Jersey

Headquarters, Newark, N. J.

President.....L. G. Huggins
1st Vice President.....G. A. Freeman
2nd Vice President.....C. E. Parmelee
Secretary.....Herbert Wolf
Assistant Secretary.....C. W. Zimmer
Treasurer.....Lester Lieberman
Board of Governors: Charles Collins, G. V. Dennis, Herbert Fox, F. C. Hawco, Lloyd Larkin, George Luce, S. H. Nitzberg, R. S. Taylor

North Piedmont

Headquarters, Greensboro, N. C.

President.....Boleslaw Jeglinski
1st Vice President.....A. J. Lane
2nd Vice President.....L. W. Welker
Secretary.....J. M. Pleasants
Treasurer.....R. D. Funderburk
Board of Governors: J. M. Daniels, L. R. Gorrell, H. W. Welker, III

Northeastern New York

Headquarters, Albany, N. Y.

President.....M. E. Waddell
1st Vice President.....L. M. Brown
2nd Vice President.....B. E. Mullen
Secretary.....G. B. Gardner
Treasurer.....E. E. Phillips
Board of Governors: E. J. Mahoney, R. D. Marshall

Northeastern Oklahoma

Headquarters, Tulsa

President.....J. E. Tumilty
Vice President.....W. C. Buckner
Secretary.....J. K. Nichols
Treasurer.....L. A. Ives
Board of Governors: C. H. Dollmeyer, G. A. McCune, R. E. Pauling

Northern Alberta

Headquarters, Edmonton, Alta., Canada

President.....R. A. Williams
Vice President.....D. E. A. MacKay
Secretary.....R. G. Proudfoot
Treasurer.....W. R. Vernon
Board of Governors: H. J. Bristow, E. K. Cumming, J. D. Henderson, E. Hodges, James Hotchkiss, E. G. McCourtney, J. Roe

Northern Connecticut

Headquarters, Hartford

President.....D. C. Allen
1st Vice President.....F. J. Raible, Jr.
2nd Vice President.....A. S. Decker
Secretary.....E. M. Johnson
Treasurer.....R. L. Miller
Board of Governors: Walter Heywood, Fritz Honerkamp

Ontario

Headquarters, Toronto, Ont., Canada

President.....R. A. Ritchie
Vice President.....Lawrence Adams
Secretary-Treasurer.....H. R. Roth
Assistant Secretary-Treasurer.....J. D. Riddle
Board of Governors: Lawrence Adams, J. D. Coates, Ernest Fox, W. A. Mould, J. D. Riddle, R. A. Ritchie, H. R. Roth, Walter Smallwood, J. W. Thompson, D. H. Wilson, W. Woodstock

Oregon

Headquarters, Portland

President.....F. T. Taylor
1st Vice President.....C. J. Bell
2nd Vice President.....W. S. Cooper
Secretary.....E. S. Constant
Treasurer.....V. C. Foster
Board of Governors: R. C. Chewning, Keith Kruchek, Ed Lokey, J. L. Waymire

Ottawa Valley

Headquarters, Ottawa, Ont., Canada

President.....A. H. Hargreaves
Vice President.....C. N. Kirby
Secretary.....G. A. Gray
Treasurer.....P. G. Fortier
Board of Governors: G. W. Goodkey, J. Klassen, J. D. Partington, G. W. Richards

Philadelphia

Headquarters, Philadelphia, Pa.

President.....J. J. Hucker
Executive Vice President.....W. F. Spiegel
1st Vice President.....W. H. Mullin
2nd Vice President.....Ludwig Mack
3rd Vice President.....O. M. Kershock
Recording Secretary.....A. A. Lincoln
Corresponding Secretary.....H. N. Teuber
Treasurer.....D. S. Plewes
Board of Governors: R. P. Anderson, C. J. Forve, J. J. Seelaus

Pittsburgh

Headquarters, Pittsburgh, Pa.

President.....Joseph Herrmann
1st Vice President.....C. W. Stanger
2nd Vice President.....D. C. Deakyne
Secretary.....William Rudoy, G. E. Smetak*
Treasurer.....R. C. Firsching
Board of Governors: E. J. Busch, H. W. Holt, J. P. Michaels, A. F. Nass, Jr.

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* Filled unexpired term.

Local Chapter Officers—1959 (*Continued*)

Puget Sound

Headquarters, Seattle, Wash.

President.....H. M. Hendrickson
1st Vice President.....M. R. Overbye
2nd Vice President.....K. G. Massart
Secretary.....J. M. Tupper
Treasurer.....D. M. Hopkins
Board of Governors: H. G. Callow, R. H. Liniger, M. W. McKinstry, W. J. Walsh, Jr., O. C. Yates

Rhode Island

Headquarters, Providence

President.....J. P. Delaney
Vice President.....C. H. Dow
Secretary.....R. E. Wilkinson, Sr.
Treasurer.....J. K. MacLean
Board of Governors: Raymond Dionne, L. M. Dunlap, Conrad Kramer, H. D. Ring, W. H. Romer, M. W. Zimmerman

Richmond

Headquarters, Richmond, Va.

President.....G. J. Wachter
Vice President.....F. J. Weiss
Secretary.....J. C. Turlington
Treasurer.....M. M. Alley
Board of Governors: H. A. Garber, G. W. Hallam, C. H. Imel

Rochester

Headquarters, Rochester, N. Y.

President.....L. C. Engelhart
1st Vice President.....H. J. Dyminski
2nd Vice President.....S. J. Stachelek
Secretary-Treasurer.....D. A. Sweetland
Board of Governors: R. E. Cherne, O. H. Heilekson, N. E. Sheldon

Rocky Mountain

Headquarters, Denver, Colo.

President.....R. J. Walker
1st Vice President.....L. R. Bindner
2nd Vice President.....J. M. Reed
Secretary.....D. Niblack
Treasurer.....R. G. Pritchard
Board of Governors: R. T. Beck, J. V. Berger, V. J. Johnson

Sacramento Valley

Headquarters, Sacramento, Calif.

President.....R. T. Andrews
Vice President.....W. B. Lander
Secretary.....L. S. Stecher
Treasurer.....R. E. Stockwell
Board of Governors: George McGrew, D. R. Peterson, J. A. White

St. Louis

Headquarters, St. Louis, Mo.

President.....E. V. Dickson
1st Vice President.....K. O. Williams
2nd Vice President.....F. G. Meyers
Secretary.....J. B. Killebrew, R. B. Tilney
Treasurer.....H. R. Halt
Board of Governors: C. F. Barmeier, J. J. Blackmore, F. G. Jaeger, C. J. R. McClure

San Diego

Headquarters, San Diego, Calif.

President.....K. N. Flocke
Vice President.....O. B. Ervin
Secretary.....K. F. Klein
Treasurer.....S. A. Bayne
Board of Governors: W. H. Hermes, III, Milton Jackson, H. A. Smith, Rex Stark, H. J. Swanson

San Joaquin

Headquarters, San Joaquin, Calif.

President.....J. P. Lamborn
Vice President.....R. C. Cody
Secretary.....D. B. Rodkin
Treasurer.....W. L. Grotzke
Board of Governors: J. A. Blayney, D. R. Lawrence

Savannah

Headquarters, Savannah, Ga.

President.....J. L. McDevitt
Vice President.....R. A. Craig, Jr.
Secretary.....S. H. Ball, Jr.
Treasurer.....J. M. Cates, E. E. Humphrey, Jr.
Board of Governors: S. H. Ball, Jr., J. M. Cates, R. A. Craig, Jr., J. L. McDevitt, R. L. Reiley

Shreveport

Headquarters, Shreveport, La.

President.....J. L. Collins, Jr.
Vice President.....J. J. Guth, Jr.
Secretary.....R. L. Johnson
Treasurer.....J. S. Tarlton
Board of Governors: L. E. Kneipp, J. M. Light, Walter Smith

South Carolina

Headquarters, Columbia

President.....J. D. Williams
Vice President.....T. O. Curlee, Jr.
Secretary.....W. O. Blackstone
Treasurer.....R. T. Waites, Jr.
Board of Governors: F. A. Bailey, III, H. J. Haar, Jr., H. B. Swygert

Local Chapter Officers—1959 (*Continued*)

South Florida

Headquarters, Miami

President.....R. D. Hazen
Vice President.....A. R. Martin, Jr.
Secretary.....Armand Cowan
Treasurer.....A. F. Plaag
Board of Governors: J. W. Hendricks, Jr., H. H. Kirkpatrick, Stephen Shelton

South Piedmont

Headquarters, Charlotte, N. C.

President.....N. W. McGuire
Vice President.....E. V. Overcash
Secretary.....W. T. Foreman, Jr.
Treasurer.....David Rickelton
Board of Governors: R. G. Agnew, E. H. Brandt, Jr., J. R. Clark

Southern Alberta

Headquarters, Calgary, Alta., Canada

President.....L. D. Ontonio
Vice President.....P. M. Meis
Secretary.....N. J. Howes
Treasurer.....W. R. Laing
Board of Governors: E. W. Deeves, J. H. Hole, O. A. Reggin, F. B. Thompson, E. H. Watson

Southern California

Headquarters, Los Angeles

President.....D. J. Missimer, Jr.
Vice President.....R. C. Taylor
Secretry.....J. C. Hall
Treasurer.....B. L. Hutchinson, Jr.
Board of Governors: C. V. Bond, R. D. Greyson, Melvin Kodmur, A. Z. Levine, K. R. Robertson

Southern Connecticut

Headquarters, New Haven

President.....R. B. Cahoon
1st Vice President.....Guy DeFeo
2nd Vice President.....Eric Olsen
Secretary.....George Dion
Treasurer.....E. F. Morley
Board of Governors: C. W. McElroy, B. M. Pactor

Toledo

Headquarters, Toledo, Ohio

President.....F. A. Edgington
Vice President.....J. E. Wilkie
Secretary.....J. S. Meyer
Treasurer.....T. F. Geikie
Board of Governors: A. L. Bolster, G. H. Frost, E. H. Williams, D. L. Wilson

Tucson

Headquarters, Tucson, Ariz.

President.....J. S. Blackmore
Vice President.....D. C. Ewald
Secretary.....H. A. Robertson
Treasurer.....J. H. Kemp
Board of Governors: A. E. Hamilton, John O'Hair, Jr., S. R. Palmer

Utah

Headquarters, Salt Lake City

President.....R. B. Oliver
Vice President.....V. Q. Tregeagle
Secretary-Treasurer.....R. N. Roser
Board of Governors: R. C. Evans, C. E. Ferguson, L. K. Irvine, Alfred Richeda

West Texas

Headquarters, Lubbock

President.....C. P. Houston
Vice President.....R. L. Mason
Secretary.....O. R. Downing
Treasurer.....J. F. Roberts
Board of Governors: W. R. Anthony, Henry Bartlett, T. F. Sartor

Western Massachusetts

Headquarters, Springfield

President.....F. J. Hurley
1st Vice President.....J. J. Curran
2nd Vice President.....Charles Martin, Jr.
Secretary.....W. K. Maki
Treasurer.....T. E. Fallon
Board of Governors: R. E. Cross, R. J. Hildreth, C. R. Munn

Western Michigan

Headquarters, Grand Rapids

President.....W. R. Johnson
1st Vice President.....S. R. Curtis
2nd Vice President.....G. L. Jepson
Secretary.....D. A. Rackliffe
Treasurer.....William Wessels
Board of Governors: R. L. Echhorn, C. H. Pesterfield, D. J. Renwick, R. J. Waalkes

Wichita

Headquarters, Wichita, Kans.

President.....F. W. Osborn
1st Vice President.....L. R. Martin, Jr.
2nd Vice President.....R. B. Peugh
Secretary.....Paul Bauer
Treasurer.....D. L. Manson
Board of Governors: Paul Bauer, M. A. Furnival, D. L. Manson, L. R. Martin, Jr., F. W. Osborn, G. L. Oswald, R. B. Peugh

Local Chapter Officers—1959 (*Continued*)

Wisconsin

Headquarters, Milwaukee

President.....R. O. Hardwick
1st Vice President.....J. E. Illingworth
2nd Vice President.....W. H. Miller
Secretary.....K. F. Waraczynski
Treasurer.....R. I. Anderson
Board of Governors: M. A. Gerke, W. I. Grant, R. D. Rodwell, A. A. Stuthelt

Special Branch

Panama and Canal Zone

Headquarters,
Curundu, Canal Zone

President.....R. L. Stephens
Secretary.....Isidro Fong

Overseas Branch

Switzerland

Headquarters, Zurich

President.....R. A. Goerg
Vice President.....A. E. Kummer
Secretary.....Kurt Wintsch
Treasurer.....H. Paerli
Headquarter.....Dr. Herbert Brown

Student Branches

North Carolina State College

Headquarters, Raleigh
Faculty Advisor.....Prof. R. B. Knight

Oregon State College

Headquarters, Corvallis
Faculty Advisor.....Prof. G. E. Thornburgh

Purdue University

Headquarters, W. Lafayette, Ind.
Faculty Advisor.....Prof. F. B. Morse

Texas A. & M. College

Headquarters, College Station
Faculty Advisor.....Prof. Billie J. Fluker

University of Detroit

Headquarters, Detroit, Mich.
Faculty Advisor.....J. B. Olivieri

University of Toronto

Headquarters, Toronto, Ont., Canada
Faculty Advisor.....Prof. F. G. Ewens

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1649

TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING REFRIGERATING AND AIR-CONDITIONING ENGINEERS

No. 1649

STORY OF CONSOLIDATION

ON JANUARY 29, 1959, the corporate consolidation between ASHAE and *ASRE* became effective. On January 26, 1959 the New York Supreme Court entered an order formally approving an Agreement for Consolidation and granting permission to file a Certificate of Consolidation. This action permitted the completion of the legal aspects.

Completion of the consolidation was the culmination of a long series of events. The object of this chapter of the TRANSACTIONS is to present an outline of these events in the form of a brief running story as shown by available records.

HISTORICAL

Projects looking toward consolidation of the two Societies were initiated on several occasions, but did not bring about a corporate consolidation. They probably were, to some extent, a causative factor in the close relations which over the years grew up between many of the organizational groups of the two Societies, as well as between society and chapter officers. There were also close professional relations between many individual members of both Societies.

The original field of interest of *The American Society of Refrigerating Engineers* was that of refrigeration, largely in its industrial aspects, while the original field of interest of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS was in connection with heating and ventilating. The growth of air conditioning tended to draw the two groups more closely together since there was overlapping of the subject matter.

As early as the 1920's there were suggestions of a consolidation. Again in the 1930's the subject was proposed by Dr. Willis H. Carrier, who was a president of both Societies. The minutes of a meeting of the ASHVE Council, held in June 1941, indicate that at that time a committee was appointed to investigate the possibilities of a closer relationship between the two Societies. It is there recorded that a Joint Committee was authorized to discuss ASHAE-*ASRE* relationships and to report back to the governing boards. *ASRE* representatives on that committee were: L. L. Lewis, Dr. W. R. Hainsworth, and H. D. Edwards, while the ASHVE representatives were, W. H. Driscoll, L. A. Harding, and W. L. Fleisher. It will be seen that the subject of a merger, or consolidation, is one which has a long background behind it.

ORIGIN OF LATEST PROJECT

There is no means of determining the actual origin of the latest and successful consolidation project. It can probably be considered to some extent to be an outgrowth of the earlier movements since it is only realistic to assume that there has always been interest in the subject on the part of many members.

In the spring of 1954 a Joint Committee on Cooperation was formed. This can perhaps be considered as the start of the latest consolidation project. Initially, though, it was not the authorized object of that committee to consider a consolidation. It is interesting to note that the minutes of the January 1955 meeting of the ASHAE Council contain a statement that "There has been no move or intent in the Joint Committee to combine the two Societies." It was not until the fall of 1955 that the Joint Committee was made an official committee of that Society by the *ASRE* Council.

Another possible source of the project is the growth of sentiment in favor of a consolidation in the local sections and chapters of the societies. Documentation of this sentiment appears in a report of the Subcommittee on Chapters Agenda Items of ASHAE, dated August 28, 1956, where it is indicated that suggestions concerning a merger were brought up from two Regions. The record further shows that these two items were referred to the Committee on Cooperation. This would indicate that by the summer of 1956 there was sentiment in favor of a possible merger in at least some of the chapters of ASHAE.

It is a matter of record that late in 1956 and during the winter of 1957 the respective Councils of the two societies did authorize the ASHAE-*ASRE* Committee on Cooperation to proceed with a study of the possibilities of a consolidation. These actions of the Councils may be regarded as marking the beginning, if not the origin, of the consolidation project.

MARCH OF EVENTS

Once the governing bodies had authorized the study by the Committee on Cooperation events moved swiftly. The committee proceeded to prepare a Plan for Merger which it was hoped would be workable and acceptable to the governing bodies and to the membership of the two Societies. Such a plan had been formulated in outline by the Joint Committee by January 1958. It was considered by the two Councils at that time. The Joint Committee worked the plan up more fully and a revised plan was ready for presentation to the Councils in June 1958. This second plan is the one which became the basis of the membership vote

at meetings of the two societies held separately on Dec. 1, 1958, and which votes were favorable and authorized the consolidation.

The period between late January 1958 and December 1, 1958 was marked by intense activity. There were committee meetings, regional meetings, chapter and section meetings. There were debates and discussions both in private and at these meetings. There were public and private statements by individuals and groups. There were press releases and editorialized statements by the presidents of both societies. The governing bodies met. Documents were authorized and prepared. There were mail communications to the members. Legal points were worked out. New By-Laws for the proposed consolidated corporation were prepared and publicized. Without doubt, although not a documentable circumstance, there was some careful thinking on the part of individual members throughout this period. In short, there were all the trappings, all the discussion and indeed all the confusion, tumult and shouting which usually marks the determination of an important decision in membership organizations where all are entitled to an opinion and to a vote.

With all the events which took place during the period when the consolidation was under active consideration, it is not practicable to attempt even a brief outline of the activities of all those who took part. Consequently the intent is to outline here only the activities of those two groups whose labors and decisions were of especial importance, *viz*: the Joint Committee and the Governing Bodies.

JOINT COMMITTEE

It was the efforts, labors, and accomplishments of this continuing committee, as constituted and re-constituted, which brought forth the Plan for Merger dated June 22, 1958 which formed the basis for the consolidation of the two societies.

The Committee was first instituted in early 1954 and was organized at its first meeting in May 1954. Although there was some alteration in its designation from time to time the Committee is generally referred to in Society documents as the ASHAE-*ASRE* Committee on Cooperation, and that is the name which appears on the flyleaf of the printed Plan for Merger dated June 22, 1958.

During the course of its operation the Committee held a series of eleven meetings as follows:

1st, New York, May 22, 1954
2nd, New York, September 25, 1954
3rd, New York, January 20, 1955
4th, New York, November 11, 1955
5th, Chicago, September 21, 1956
6th, Boston, November 25, 1956
7th, Chicago, February 26, 1957
8th, Chicago, December 6, 1957
9th, Pittsburgh, January 28, 1958
10th, Chicago, April 18, 1958
11th, Chicago, May 18, 1958

Minutes report the events which took place at these meetings. In addition to these meetings, since the Committee operated with subcommittees, there were numerous meetings of these smaller groups. Generally, a good idea of reports and findings of these groups can be obtained from documents attached to the minutes of the major meetings. The first five meetings of the Committee—those held during 1954, 1955 and September 1956—were devoted to studies of many phases of the activities of the two societies with a view of determining in what ways the two societies could co-operate without a consolidation taking place. The last

six meetings—from November 1956 to May 1958—were concerned largely with the subject of a consolidation of the two societies.

As originally constituted, the Committee was composed of five representatives of each society. This continued through the period when the first four meetings were held (1954 through 1955). Serving as *ASRE* representatives during this period were F. H. Faust, C. F. Kayan, James Larkin, J. F. Stone and N. N. Wolpert during the first two meetings; F. H. Faust, A. J. Hess, C. F. Kayan, J. F. Stone and M. C. Turpin during the third and fourth meetings. At this same time the ASHAE representatives were M. F. Blankin, P. B. Gordon, J. E. Haines, E. N. McDonnell and E. R. Queer during the period of the first three meetings; and M. F. Blankin, P. B. Gordon, L. N. Hunter, E. N. McDonnell and E. R. Queer during the fourth meeting. Throughout the period covered by the first four meetings, F. H. Faust served as chairman of the Committee and M. C. Turpin served as committee secretary. By the time of the fifth meeting (September 1956) the Committee had been re-constituted to consist of three members from each society. The representatives of *ASRE* were A. J. Hess, Cecil Boling and H. F. Spoehrer, while those of ASHAE were L. N. Hunter, E. R. Queer and Cary Gamble. At the fifth meeting, A. J. Hess was elected chairman and R. C. Cross was elected secretary of the Committee. These representatives and officers continued without change through the ninth meeting in January 1958. By the time of the 10th meeting the representatives of each society had been increased to five with the addition of E. N. Johnson and E. J. Tarleton of *ASRE*, and of W. G. Hole and P. J. Marschall of ASHAE.

The early meetings of the Committee brought forth several reports to the Councils of the two societies but these were not directly related to the subject of a consolidation. The chief documents appearing as a result of the work of the committee and dealing directly with the subject of the consolidation are:

(A) REPORT ASHAE-*ASRE* COMMITTEE ON COOPERATION TO ASHAE AND *ASRE* COUNCILS—JANUARY 26, 1958

(B) PLAN FOR MERGER—ASHAE AND *ASRE* (Approved in principle by the Councils, JUNE 22, 1958).

ACTIONS OF THE GOVERNING BODIES

The governing boards of the two societies (*ASRE* Council and ASHAE Council) devoted many hours of the closest attention to the consolidation project from its inception to its completion.

At the ASHAE Council meeting April 4, 1954, President Hunter announced the names of the ASHAE representatives and at its July 11, 1954 meeting President Hess similarly announced to the *ASRE* Council the names of the *ASRE* representatives and reported on the highlights of the first meeting of the Joint Committee. The July 26, 1954 meeting of the ASHAE Council heard a report of the same Joint Committee meeting presented by Chairman Faust.

At the December 1954 and January 1955 meetings of the respective governing bodies, the Joint Committee report covering the activities of that Committee including its meeting of September 25, 1954 was presented.

At its November 30, 1955 meeting the *ASRE* Council voted that the Joint Committee and all its subcommittees be affirmed and made a Joint Committee of the Society.

By the fall of 1956 the Joint Committee had made its recommendation that the Committee be re-constituted with its membership reduced from 5 representatives of each Society to three each, and that its scope and purpose be restated. Accordingly, at the November 26, 1956 meeting of the *ASRE* Council and at the February 24, 1957 meeting of the ASHAE Council, the re-constitution of that committee was authorized. The *ASRE* Council voted to the committee the job of studying the merits and demerits of merger with any other society, and limited the study to an exploration. The ASHAE Council voted that the scope of the committee be broadened to include a factual study of the percentage of membership interested in a merger and to explore possible plans for future consideration.

On January 26, 1958 the governing bodies met separately in Pittsburgh. The ASHAE Council had previously received copies of the report of the Joint Committee on Cooperation dated January 26, 1958. At this meeting the following resolution was

> **VOTED:** THAT the ASHAE-*ASRE* Plan dated January 26, 1958, be presented to the membership of ASHAE through the Chapters Regional Committee meetings held during the spring of 1958, contingent upon similar action by the Council of *ASRE*, and
>
> **THAT** the ASHAE Committee members be authorized to proceed with further detailed study on the proposed merger of the two Societies.

At the meeting of the *ASRE* Council on January 26, 1958 the Report of the Joint Committee dated January 26, 1958 was received and the Council unanimously voted approval in principle, of the method of merger set forth in that report, and approved the continuance of the Program of the Joint Committee, and also voted unanimously to request the Joint Committee to report to the Council further details of the method of merger at the meeting of the Council in June 1958, and that if approved by the Council, it be submitted to the 54th Annual Meeting in June 1958, for membership approval for submission at the Semi-Annual Meeting on December 1, 1958.

Each of the governing bodies at their January 26, 1958 meetings authorized the issuance of a joint press release at that time.

At the April 20, 1958 meeting of ASHAE Council, L. N. Hunter reported on the Joint Committee meeting held on April 18, 1958, and the Council voted that W. G. Hole and P. J. Marschall be made members of the Joint Committee. Mr. Hunter also reported that subcommittees of the Joint Committee had held meetings concerning research, standards, publications, and program and papers in order to include recommendations concerning these activities in its report. President Queer announced that he had sent a letter to all chapters in February.

By June 22, 1958 the Joint Committee had completed its "Plan for Merger" bearing that date, and both of the governing bodies met separately that same day.

At the ASHAE Council meeting the following resolutions, among others, were voted.

> **VOTED:** THAT the Council approve in principle the plan for merger with *ASRE*, as set forth in the memorandum of the ASHAE-*ASRE* Committee on Cooperation dated June 22, 1958.

> **VOTED:** THAT the Executive Committee be empowered to take such steps as are proper and expedient to prepare such documents as are appropriate for the consolidation of ASHAE and *ASRE* to be presented to the October 1958 meeting of the Council.

At its meeting held June 22, 1958 President Spoehrer reported to the *ASRE* Council that he had appointed E. N. Johnson and F. L. Tarleton to become members of the Joint Committee. The Council ratified these appointments. The Council voted its approval of the following recommendation of the Executive Committee of *ASRE*:

> The Executive Committee recommends to Council the acceptance, in principle, of the plan of merger of *ASRE* and ASHAE as set forth in the Second Report to the Council of the *ASRE*-ASHAE Committee on Cooperation, dated June 22, 1958, and further recommends that the Council refer the plan to the membership of the Society at the meeting of the Society on Monday, June 23, 1958, for authorization to submit the plan to the membership for consideration and vote at the Semi-Annual Meeting of the Society in December, 1958.

At the June 28, 1958 meeting of the *ASRE* Council President Boling reported that a meeting of the combined executive committees of the two societies, along with the Committee on Cooperation had been held the day before and that a tentative time table had been set up. It was indicated that it was expected that in October the entire official plan could be sent to the voting members of the Society. A news release was prepared.

At its meeting held October 18, 1958 the ASHAE Council approved the form and wording of the Announcement to ASHAE Members and of the Proxy Form and Meeting Notice. Approval of the Agreement for Consolidation was voted by the following resolution:

> **RESOLVED** THAT the proposed agreement for consolidation be and the same hereby is in all respects approved, and further resolved
>
> **THAT** the appropriate officers of the Society be and hereby are authorized and directed to make, execute and deliver said agreement.

The Council also voted that the executive committee be directed to work toward the amendment of various details in the wording of the proposed By-Laws for the merged Society.

CONCLUDING EVENTS

By early August, 1958 the Plan for Merger was in readiness for submission to members for study. Under date of August 11, 1958 a "Summary of Proposed Merger Plan," with an accompanying transmittal letter by *ASRE* President Boling and a similar letter by ASHAE President Queer, was mailed to the members of the two Societies. Under date of August 20, 1958 President Queer by a letter to all chapter presidents and secretaries directed that the Plan of Merger be discussed in the September, October or November meetings of ASHAE chapters, and that Regional Directors be informed of chapter plans concerning the schedule of this discussion.

Under date of Oct. 22, 1958 President Queer and Executive Secretary Hutchinson mailed an "Announcement to all ASHAE Members on the Merger," enclosing a notice of the Special Meeting of Dec. 1, 1958, a proxy form for use in voting at that meeting, and the Agreement for Consolidation. This announcement, the meeting notice, and the proxy form are printed in full in Chapter 1650 of this volume of the TRANSACTIONS.

Similarly, under date of October 22, 1958, President Boling addressed a letter to *ASRE* members under the head THE TIME HAS COME—TO DECIDE—TO CAST YOUR VOTE. To the letter was attached a form for proxy voting at the *ASRE* 45th Semi-Annual Meeting on December 1, 1958. Enclosed also was a copy of the Agreement for Consolidation and the NOTICE OF MEETING dated October 22, 1958 and signed by President Boling and Executive Secretary Cross.

Voting by the membership of *ASRE* on the consolidation proposal took place at the opening session of the 45th Semi-Annual Meeting held in New Orleans on Monday, December 1, 1958. In *Refrigerating Engineering* (January 1959, p. 50) it is stated that votes were cast by 86 percent of the members qualified to vote, and that the result was that there were 3516 favorable votes with 1293 unfavorable.

Voting by the membership of ASHAE took place at a special meeting called for the purpose in Chicago on December 1, 1958. The result of that vote, as stated in a letter dated December 2, 1958 by President Queer and sent to all ASHAE members, was 5307 favorable votes and 405 unfavorable.

Plans for the 65th Annual Meeting of ASHAE in Philadelphia January 26–29, 1959 were carried out although the result of the voting had assured that a consolidation of the two societies would take place as soon as the necessary legal formalities were accomplished. At that meeting the officers and members of the ASHAE Council already elected were installed in office. These officers were: Arthur J. Hess, Los Angeles, Calif., President; Walter A. Grant, Syracuse, N. Y., 1st Vice President; John Everetts, Jr., Philadelphia, Pa., 2nd Vice President; and John H. Fox, Toronto, Ont., Canada, Treasurer. The newly elected members of the Council were William J. Collins, Jr., Oklahoma City, Okla.; Harold A. Lockhart, Morton Grove, Ill.; Rolland S. Stover, Marshalltown, Ia.; and George W. F. Myers, St. Louis, Mo. Those just named are therefore recorded as the officers and newly installed members of Council of ASHAE from the evening of January 28, 1959 until the consolidation became effective on January 29, 1959.

On the morning of January 29, 1959, the needed documents were filed in the office of the Secretary of State in Albany. Thus in its 55th year the *American Society of Refrigerating Engineers* ceased its corporate existence, and in its 65th year the charter of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS was amended to bring into corporate existence the consolidated AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC. The initial officers of the consolidated Society were: President, Cecil Boling; 1st Vice President, A. J. Hess; 2nd Vice President, D. D. Wile; 3rd Vice President, Walter A. Grant; 4th Vice President, R. H. Tull; 5th Vice President, John Everetts, Jr.; 1st Treasurer, J. H. Fox; 2nd Treasurer, F. Y. Carter; Executive Secretary, A. V. Hutchinson; while the membership of the initial Board of Directors comprised those listed in the Agreement for Consolidation on page 31 of this volume of TRANSACTIONS.

Operation of ASHRAE began with a meeting of its Executive Committee at 10:30 a.m. in the Bellevue Stratford Hotel in Philadelphia, Pa., on January 29, 1959.

BIBLIOGRAPHY

Joint Press Releases—One dated January 28, 1958 can be found in full on p. 179, ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, March 1958. The second, dated July 1, 1958 in full on p. 163, ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, August 1958.

President's Pages in *Refrigerating Engineering*—p. 51, March 1958; pp. 50 and 51, June 1958; p. 38, August 1958; p. 41, November 1958; pp. 48 and 49, January 1959.

Discussion Pages in *Refrigerating Engineering*—p. 64, July 1958; pp. 46 and 47, August 1958.

President's Message pages in ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, 1958: p. 146, March; p. 138, June; p. 112, July; p. 128, August; p. 154, September; p. 110, October.

1650

No. 1650

SPECIAL MEETING DECEMBER 1, 1958

CHICAGO, ILL.

A SPECIAL MEETING of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS was held at 10:30 a.m. in the Beverly Room of the Conrad Hilton Hotel, Chicago, Ill., on December 1, 1958. Notice of this meeting* was mailed on or about October 22, 1958 to all members of the Society eligible to vote. Appendix A is a reprinting of the notice of this meeting, which was mailed with a letter signed by Pres. E. R. Queer, University Park, Pa., and Executive Secy. A. V. Hutchinson, New York, N.Y., dated October 22, 1958, and reprinted here as Appendix B. Referred to in item 2 of Appendix B is the Proxy Form which is reprinted here as Appendix C.

This Special Meeting was called to order by President Queer, who, on noting that a quorum was present, recognized First Vice Pres. A. J. Hess, Los Angeles, Calif., who proposed the following resolutions concerning the proposed merger:

> RESOLVED THAT the proposed Agreement for Consolidation, dated as of October 18, 1958, be and same hereby is in all respects ratified, confirmed and approved; and further
>
> RESOLVED THAT the appropriate officers of the Society be and they and each of them are hereby authorized and directed to make, execute and deliver said Agreement for Consolidation, file the Certificate of Consolidation and take any and all further actions and proceedings as may be necessary or desirable to effect the consolidation of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC. and *The American Society of Refrigerating Engineers*.

The resolution was seconded by G. B. Priester, Baltimore, Md.

President Queer then announced the appointment of the Inspectors of Election as follows: Walter A. Grant, Syracuse, N.Y., as chief inspector; S. R. Lewis, Chicago, Ill.; and John Everetts, Jr., Philadelphia, Pa.

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* It will be observed that the notice of the meeting (Appendix A, herewith), in the third paragraph, mentions an Agreement of Consolidation, a copy of which was annexed to each copy of the meeting notice mailed to members, and made a part of the meeting notice by reference, and marked *Exhibit I* in the meeting notice. The material contained in *Exhibit I* is the same material printed under the head "I—Agreement for Consolidation" in Chapter 1651 of this volume of TRANSACTIONS. It begins on p. 13 and continues through to p. 33.

President Queer also stated that any member present who wished to change his vote or withdraw his proxy could do so until 11:15 a.m., at which time the ballot box would close.

With the floor open for discussion of the proposed resolutions concerning the merger, J. H. Aeberly, Chicago, Ill., made a statement explaining why he was voting against the resolutions.

President Queer then announced that he would hold the meeting open long enough to permit the announcement of the result of the proxy vote following the closing of the ballot box at 11:15 a.m.

As there was no further discussion concerning the merger resolutions, President Queer then announced that the next order of business would be that of giving notice of the proposed amendment to the By-laws changing the time of the Chapters Regional Committee meetings as included in the meeting notice. President Queer read the proposed amendment to **Article VII,** *Section 3 (f)*, next to the last sentence, which would read

"... The said committees shall hold committee meetings prior to December 1 of each year."

The wording of the sentence in the present By-laws is:

"... The said committees shall hold committee meetings prior to June 1 of each year."

President Queer explained that the action in presenting the change at this Special Meeting is to permit the change to be voted on by the membership at the next Annual Meeting of the Society, and that no vote on it is in order at the Special Meeting.

Discussion concerning the proposed merger took place with G. L. Tuve, Cleveland, Ohio, asking about meeting plans if the proposed merger resolutions are favorably voted, and about the administration of research. Herbert Kreisman, Chicago, Ill., asked for information about problems of local chapters, especially as regards date of consolidation and disposition of the funds of local chapters. Other points were raised by L. E. Seeley, Westfield, Mass., and E. R. Teske, Chicago, Ill. Others taking part in the discussion included First Vice President Hess, L. N. Hunter, Johnstown, Pa., B. H. Jennings, Cleveland, Ohio, and Mr. Everetts.

At 11:15 a.m., Chief Inspector of Election Grant announced that the ballot box was closed. S. R. Lewis and John Everetts, Jr. then cast the ballots on the resolutions as presented. A total of 5727 ballots had been received, of which 15 were found to be invalid, leaving a total of 5712 valid ballots. The total number of valid ballots cast in favor of the resolutions was 5307, or 93 percent; while the total cast against the resolutions was 405, or 7 percent.

President Queer then announced that the ballot showed that the members had approved the resolutions, and adjourned the meeting at 11:20 a.m.

APPENDIX A

NOTICE OF SPECIAL MEETING TO ALL MEMBERS ENTITLED TO VOTE

NOTICE IS HEREBY GIVEN that a Special Meeting of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., will be held at the Conrad Hilton Hotel, 720 South Michigan Ave., Chicago, Ill., on December 1, 1958, at 10:00 o'clock A.M. CENTRAL STANDARD TIME.

The objects of the meeting are (A) to vote upon the approval of the Agreement for Consolidation providing for the merger of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and *The American Society of Refrigerating Engineers;* (B) to authorize and empower the officers and directors to join with the officers and directors of *The American Society of Refrigerating Engineers* in making, executing, and filing a Certificate of Consolidation, and to take any and all further actions and proceedings necessary or desirable to effect the consolidation.

A copy of the Agreement for Consolidation is annexed hereto, made part hereof, and marked *Exhibit I.*

Notice of a proposed amendment to the By-Laws changing the time of the Chapters Regional Committee Meetings will be given to the members at said Special Meeting.

The approval of the Agreement for Consolidation requires the affirmative votes of two-thirds of the members of each of the Constituent Societies entitled to vote, present, in person or by proxy, at the respective meetings called for such purpose.

A Proxy is attached containing spaces for write in names, and you may signify thereon how you wish your vote to be cast. The Proxy will be voted in accordance with your instructions; but if no instructions are given, it will be voted in favor of the consolidation.

Dated, New York, N. Y., October 18, 1958. By order of the Council.
E. R. QUEER, *President* A. V. HUTCHINSON, *Secretary*

APPENDIX B

ANNOUNCEMENT TO ALL ASHAE MEMBERS ON THE MERGER

Every member of the ASHAE is requested to read the documents enclosed with this statement:

1. The call for a Special Meeting of ASHAE on December 1, 1958 at 10 a.m. C. S. T. in the Conrad Hilton Hotel, Chicago, Ill., for the purpose of voting on the consolidation of this Society and *The American Society of Refrigerating Engineers*—the merged Society to be called: AMERICAN SOCIETY OF HEATING REFRIGERATING AND AIR-CONDITIONING ENGINEERS.
2. The Proxy Form to be used for voting in favor of or against the Merger. If you wish to have your vote recorded, mark and sign the Froxy then mail it in the enclosed return envelope.
3. The Officers and the Council unanimously urge you to vote in favor of the Merger as they believe:
   (*a*) that it is desirable and in the best interest of the members to merge ASHAE and *ASRE.*
   (*b*) The consolidation will expand the scope, prestige and effective service of both organizations.
   (*c*) The enlarged Society will be able to extend its programs, sponsor better research programs, expand its code and standards activities, enlarge its education programs through better meetings and publications.
   (*d*) A combined Society offers many professional advantages to the individual members as well as to the regional and local chapter groups.
4. The proposed By-Laws should be carefully studied. In the opinion of the Officers and the Joint Committee on Cooperation, they combine the most desirable features of the ASHAE and *ASRE* and should result in substantial economies in the administration and operation of a consolidated Society over two independent groups.

In the summary of the Proposed Merger Plan dated August 11, 1958 and mailed to all members by the respective Presidents, Messrs. E. R. Queer, ASHAE and Cecil Boling, *ASRE*, an outline of the plan and its operation were furnished for study. This is now supplemented by the enclosed (*a*) Agreement for Consolidation which gives the timetable governing the plan, the new name, the powers and objectives in full detail and the procedure of filing a Certificate of Consolidation in the event that the merger plan vote is formally completed at the Special Meeting of ASHAE December 1, 1958 at the Conrad Hilton Hotel, Chicago, Ill. and at the Semi-Annual Meeting of *ASRE* on December 1, 1958 at the Roosevelt Hotel, New Orleans, La.

Approval of the merger may be effected by the votes cast by two-thirds of the members of each of the constituent Societies entitled to vote, present, in person or by proxy at the respective meetings called for such purpose. Therefore, it is the responsibility of every member eligible to vote to express his wish on this vital matter.

E. R. QUEER, *President*

October 22, 1958 A. V. HUTCHINSON, *Secretary*

APPENDIX C

PROXY

I hereby appoint P. B. Gordon, L. N. Hunter, S. R. Lewis, L. E. Seeley or G. L. Tuve, or_____, or_____, as my proxy or proxies, to vote in my behalf at a Special Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., to be held at the Conrad Hilton Hotel, Chicago, Ill., on December 1, 1958, at 10 a.m., C.S.T., or any adjournment or adjournments thereof; and I direct that my said proxies shall vote as indicated by the following markings:

☐ In favor of the resolutions for the consolidation of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and *The American Society of Refrigerating Engineers*.

☐ Against the resolutions for consolidation.

If no markings appear, my proxies are hereby authorized to cast my vote in *favor* of the resolutions for the consolidation.

I do hereby further authorize my said proxies to vote in my behalf according to their discretion upon any other relevant matters which may properly come before said Special Meeting.

Dated, _____, 1958 _____

(Signature)

1651

No. 1651

CONSOLIDATION DOCUMENTS

TO MEET the requirements of the Membership Corporations Law of the State of New York a number of documents had to be prepared in order to bring about the consolidation of ASHAE and *ASRE*. These several documents are printed in full in this chapter in order that they may become a part of the TRANSACTIONS record.

	PAGE
I—AGREEMENT FOR CONSOLIDATION	13
Exhibit A—Consolidated By-Laws	18
Exhibit B—Officers and Board of Directors (January 1959 to June 1962)	31
II—PETITION FOR CONSOLIDATION	33
Exhibit A—Provisions of Certificate of Incorporation of ASHAE	36
Exhibit B—Provisions of Certificate of Incorporation of *ASRE*	38
III—ORDER OF COURT FOR CONSOLIDATION	39
Affidavit, and Consent of Education Department	41
IV—CERTIFICATE OF CONSOLIDATION	42

The printing of the foregoing mentioned documents is in accordance with a resolution of the Executive Committee of the consolidated Society at its organization meeting held on January 29, 1959. The resolution is as follows:

RESOLVED THAT a certified copy of the Certificate of Consolidation, and true copies of the foregoing order and consents, the joint petition for consolidation, the provisions of the respective certificates of incorporation of the Constituent Societies, the Agreement for Consolidation and the exhibits attached thereto, and the affidavit of Louis P. Eisner submitted in support of the petition, be filed in the minute book of the consolidated Society and attached therein as permanent exhibits, and that all the said documents and papers be published in the Society's TRANSACTIONS.

I—Agreement For Consolidation*

Agreement for consolidation made and entered into as of the 18th day of October, 1958, by and between AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING

* This Agreement for Consolidation, including Exhibits A and B, was annexed to the Notice of Special Meeting of ASHAE, dated October 18, 1958, and which notice is shown in full in Chapter No. 1650 of this volume of TRANSACTIONS. It was also enclosed with the letter which President Boling addressed to *ASRE* members under date of October 22, 1958. For the information and consideration of members it was printed on both sides of a 14- × 25-in. blue sheet, and on which it was designated "Exhibit I."

ENGINEERS, INC., hereinafter sometimes called ASHAE, having its principal office at 62 Worth St., New York 13, N. Y., and *The American Society of Refrigerating Engineers*, hereinafter sometimes called *ASRE*, having its principal office at 234 Fifth Ave., New York 1, N. Y., both of said parties being hereinafter sometimes jointly referred to as the "Constituent Societies."

WHEREAS, ASHAE was incorporated on January 24, 1895, and is a corporation duly organized and existing under the Membership Corporations Law of the State of New York; and

WHEREAS, *ASRE* was incorporated on August 30, 1905, and is a corporation duly organized and existing under the Membership Corporations Law of the State of New York; and

WHEREAS, the boards of directors of the Constituent Societies, after due deliberation, have resolved that it is desirable and in the best interests of said corporations and the respective members thereof that said corporations shall be consolidated under and pursuant to the laws of the State of New York, and that the public interest would not be adversely affected by the consolidation of said corporations; and

WHEREAS, it is the intention of the parties that the combined purposes and activities of the Constituent Societies shall be continued by means of the consolidation, with such changes as may from time to time be deemed advisable by the members and by the Board of Directors of the consolidated corporation, to the end that from the enlarged activities and services of the consolidated corporation the members and the public may derive greater benefits and the general welfare advanced.

NOW, THEREFORE, IN CONSIDERATION of the premises and of the mutual agreements, covenants, and provisions herein contained, and for the purpose of prescribing the terms and conditions of the consolidation and setting forth the mode of carrying the same into effect, the parties hereto do hereby covenant and agree as follows:

1. This Agreement for Consolidation shall be submitted for approval to the members of each of the Constituent Societies entitled to vote, at separate meetings thereof, to be duly held prior to December 31, 1958, upon due notice; that as promptly as may be practicable after due approval of this agreement by the members of each of the Constituent Societies and their authorization to make, execute, and file a Certificate of Consolidation substantially in the form hereinafter set forth, the parties hereto shall take appropriate action and proceedings to effect such consolidation; they shall jointly apply to the Supreme Court of the State of New York for an order approving this agreement and authorizing the filing of the Certificate of Consolidation; and following the making of such an order, and obtaining such other consents and approvals as may be necessary, shall duly file the Certificate of Consolidation in the office of the Secretary of State of the State of New York.

2. The Certificate of Consolidation shall be in the form prescribed by *Section* 50 of the Membership Corporations Law and, in addition to the formal and other provisions required by said section, by governmental departments, agencies, or officials, and by the court, said certificate shall include clauses reading substantially as follows:

I. The name of the consolidated corporation is AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

II. The territory in which its operations are principally to be conducted is in all parts of the United States, its territories and possessions, in the Dominion of Canada, and in any and all foreign countries and places.

III. The office of the consolidated corporation is to be located in the City, County, and State of New York.

IV. The number of its directors shall be not less than twenty-four (24) nor more than forty-eight (48).

The directors and elective officers, and their respective terms of office, shall initially be as set forth in the agreement for consolidation, and thereafter the directors and elective officers, and their respective terms of office, shall be as set forth in the by-laws of the consolidated corporation.

In furtherance and not in limitation of the powers conferred by statute, the board of directors, if the by-laws so provide, may designate one or more committees which, to the extent provided in the by-laws of the consolidated corporation, or by resolution or resolutions of the board of directors, shall have and may exercise the powers of the board in the management of the business and affairs of the consolidated corporation that may be delegated to such committee or committees, such committee or committees to

have such name or names as may be stated in the by-laws, or as may be determined from time to time by resolution or resolutions of the board of directors.

The consolidated corporation may in its by-laws confer powers and authorities upon its board of directors in addition to the powers and authorities expressly conferred by the statutes of the State of New York or by this certificate. All powers of the consolidated corporation, insofar as the same may be lawfully vested in the board of directors, are hereby conferred upon the board of directors of the consolidated corporation.

Both members and directors of the consolidated corporation shall have the power to hold their respective meetings within or without the State of New York; the consolidated corporation shall have the power to maintain offices and keep the books of the consolidated corporation within or without the State of New York, at such place or places as may from time to time be designated by the board of directors.

No officer, director, or member of the consolidated corporation shall receive or be lawfully entitled to receive any part of the net earnings thereof or any pecuniary profit from the operations thereof, except such reasonable compensation for services in effecting one or more of its purposes as the board of directors may determine.

Each director of the consolidated corporation shall be indemnified by said corporation against expenses actually and necessarily incurred by him in connection with the defense of any action, suit, or proceeding in which he is made a party by reason of his being or having been a director of the corporation, except in relation to matters as to which he shall be adjudged in such action, suit, or proceeding to be liable for willful negligence, misfeasance, or misconduct in the performance of his duties as director; such right of indemnification shall not be deemed exclusive of any other right to which he may be entitled under any by-law, agreement, vote, or otherwise

V. The consolidated corporation shall be a new corporation, or either one of the Constituent Societies.

VI. The terms and conditions of consolidation, in addition to those hereinabove set forth, and the mode of carrying the same into effect, are as follows:

(1.) The purposes of the consolidated corporation are:

(a) To advance the arts and sciences of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences, for the benefit of the general public.

(b) To encourage and conduct scientific research and the study of principles and methods in the fields of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences, the results of which shall be made freely available to the public.

(c) To promote the unrestricted dissemination of knowledge and information and, for such purpose, to publish and to foster the publication of books, periodicals, reports, educational programs, and scientific and technical data relating to heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences.

(d) To engage in educational activities (not including the conduct of any school or institution of learning), and to encourage the adoption and maintenance of high standards of instruction and technical and professional training in the fields of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences.

(e) To cooperate with governmental agencies and with universities, colleges, schools, and other organizations and groups having the same or similar objects and purposes, and to establish scholarships and make contributions, grants, and awards in furtherance of the foregoing purposes.

(f) To establish, approve, adopt, and promulgate codes, standards, and procedures in the fields of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences, subject to the proviso that all such activities shall be conducted solely for the advancement of engineering science.

(g) To organize and form local regions, chapters, sections, and student and other branches, and to regulate, operate, and control the same under the direction and at the pleasure of the consolidated corporation, but no local region, chapter, section, or branch shall subject the consolidated corporation to any financial or other obligation except such as the consolidated corporation may voluntarily assume.

(h) To receive, acquire, hold, and maintain any property, real or personal, without limitation as to amount or value, for any of the consolidated corporation's objects, by way of bequest, devise, gift, grant, purchase, or lease, to invest and reinvest the same, to control the income therefrom, and to expend or otherwise dispose of all or any portion of its funds and property, including the income, interest, or principal, subject, however, to any directions or limitations placed upon the same by donors or testators.

(i) To do any and all things necessary or proper in connection with or incidental to any of the foregoing.

(j) The consolidated corporation shall be operated exclusively for scientific and educational purposes; no substantial part of the activities of the consolidated corporation shall be the carrying on of propaganda or otherwise influencing or intending to influence legislation; in the event of the dissolution of the consolidated corporation, the board of directors shall dispose of its net assets, in trust, however, to further the purposes expressed herein, without preference in favor of any contributor or any member, officer, or director of the consolidated corporation, and subject to the order of a Justice of the Supreme Court.

(2.) Upon the filing of the Certificate of Consolidation in the office of the Secretary of State, each member of the constituent corporations shall have and possess the highest grade of membership in the consolidated corporation as was upon such filing held by him in either of the constituent corporations; and each member of the constituent corporations then entitled to vote therein shall possess the power to vote in the consolidated corporation for the election of directors and for all other purposes conferred upon voting members by the laws of the State of New York, this Certificate of Consolidation, and the by-laws.

(3.) Except as provided in subdivision 5 of this Paragraph VI, upon the filing of the certificate of consolidation in the office of the Secretary of State, the separate existence of each constituent corpora-

tion, except the consolidated corporation shall cease and the consolidation shall be effective, as provided in *Section* 53 of the Membership Corporations Law and the consolidated corporation shall possess all of the powers of the constituent corporations, and shall have the power and be subject to the duties and obligations of a membership corporation formed under the laws of the State of New York for like purposes.

(4.) Upon the filing of the Certificate of Consolidation in the office of the Secretary of State, all the rights, privileges, and interests of each of the constituent corporations, all the property, real, personal, and mixed, and all the debts due on whatever account to either of them, and other things in action belonging to either of them, shall be deemed to be transferred to and vested in the consolidated corporation without further act or deed, and all claims, demands, property, and every other interest, whether vested or contingent, shall be as effectually the property of the consolidated corporation as they were of the constituent corporations, and the title to all real estate, taken by deed or otherwise under the laws of the State of New York vested in either of the constituent corporations, shall not be deemed to revert or be in any way impaired by reason of the consolidation but shall be vested in the consolidated corporation.

(5.) Except as provided by *Section* 12 of the Personal Property Law, or *Section* 113 of the Real Property Law, any devise, bequest, gift, or grant contained in the will of a person dying domiciled in the State of New York, or in any other instrument executed under the laws of said State, taking effect after such consolidation, to or for either of the constituent corporations, shall inure to the benefit of the consolidated corporation; and so far as is necessary for that purpose, or for the purpose of a like result with respect to a devise, bequest, gift, or grant governed by the law of any other jurisdiction, the existence of each constituent corporation shall be deemed to continue in and through the consolidated corporation.

(6.) Upon the filing of the Certificate of Consolidation in the office of the Secretary of State, the consolidated corporation shall be deemed to have assumed and shall be liable for all the liabilities and obligations of the constituent corporations, in the same manner as if such consolidated corporation had itself incurred such liabilities or obligations.

Until the date of the filing of said Certificate of Consolidation in the office of said Secretary of State, the directors of the Constituent Societies are hereby vested with the power and discretion, by a two-thirds vote of the directors of each of the Constituent Societies present at a meeting, to make editorial changes and changes in language and form of the Certificate of Consolidation, subject, however, to the proviso that no such change shall alter the substance of the foregoing.

3. Upon the filing of said Certificate of Consolidation, the By-laws of the consolidated corporation shall be as set forth in *Exhibit A*, annexed hereto and made a part hereof. From and after the effective date of the consolidation and until said *Exhibit A* shall be altered, amended, or repealed, or new By-laws duly adopted, said *Exhibit A* shall be the By-laws of the consolidated corporation. Until the filing of the Certificate of Consolidation, the Boards of Directors of the Constituent Societies may vary the By-laws set forth in *Exhibit A* to conform with the requirements of governmental departments, agencies, or officials, or the court; and until December 31, 1959, the directors of the consolidated corporation shall be vested with the power and discretion, by a two-thirds vote of the directors present at a meeting, to fill blanks and make editorial changes and changes in the language and form of *Exhibit A*, subject, however, to the proviso that such changes by the directors of the consolidated corporation shall not alter the substance of *Exhibit A*.

4. On the effective date of the consolidation and notwithstanding any of the provisions of *Exhibit A*, the number of directors of the consolidated corporation shall be forty-seven (47); at the close of the Annual Meeting of the consolidated corporation to be held during the summer of 1959, the said number shall automatically be decreased to thirty-four (34); at the close of the Semi-Annual Meeting to be held during the winter of 1960, the said number shall automatically be decreased to twenty-eight (28); at the close of the Semi-Annual Meeting to be held during the winter of 1961, the said number shall automatically be decreased to twenty-seven (27); at the close of the Annual Meeting to be held during the summer of 1961, the said number shall automatically be increased to twenty-eight (28); at the close of the Annual Meeting to be held during the summer of 1962 the number shall automatically be decreased to twenty-four (24); after said 1962 Annual Meeting the number of directors may from time to time be increased or decreased in the manner permitted by the statutes of the State of New York, the Certificate of Consolidation, and *Exhibit A*.

5. Notwithstanding any of the provisions of the Certificate of Consolidation and *Exhibit A*, the officers and directors of the consolidated corporation, and the members of the Executive Committee, without further action or vote of the members of the consolidated corporation, shall be the persons indicated in *Exhibit B*, annexed hereto and made a part hereof. Each of the persons named in said *Exhibit B* shall hold the office and committee membership assigned to him therein until the date of termination set over his name in said *Exhibit B* and until his successor shall have qualified, or until his death, or until he shall have resigned or shall have been removed from office in accordance with law; if, on the effective date of the consolidation, or during the term of office specified for such persons in *Exhibit B*, a vacancy or vacancies shall occur in any directorship, office, or committee membership, such vacancy or vacancies shall be filled by the Board of Directors then in office for the balance of the term prescribed for such vacancy or vacancies in said *Exhibit B*. Where no names are specifically designated in said *Exhibit B*, the directorships and offices involved shall be filled by the members in accordance with the nominating and election procedures prescribed in *Exhibit A* and vacancies therein shall be filled in accordance therewith.

6. The assets and liabilities of the Constituent Societies shall be taken upon the books of the consolidated corporation as at the close of business on the effective date of the consolidation, at the amounts at which they shall on said date be carried on the respective books of the Constituent Societies. However, the assets and liabilities of the Chapters, Sections, and Branches of the Constituent Societies shall not be entered but the net assets thereof shall be transferred and assigned to trustees or committees of each such Chapter, Section, and Branch, and shall be used by such trustees or committees to further one or more of the purposes of the consolidated corporation as expressed in the Certificate of Consolidation.

7. Prior to the effective date of the consolidation neither of the Constituent Societies shall authorize, approve, or enter into any transaction or transactions other than in the ordinary course of business, except with the consent and approval of a majority of the persons designated in *Exhibit B* as the original directors of the consolidated corporation.

8. If at any time either of the Constituent Societies or the consolidated corporation shall determine or be advised that any further resolutions, assignments, assurances in law, or other things are necessary or desirable to vest in the consolidated corporation according to the terms of this agreement the right, title, and interest to any property of the Constituent Societies, each of the Constituent Societies, and the proper officers and directors thereof, shall and will execute and deliver any and all such proper resolutions, assignments, assurances in law, or things necessary or desirable to vest title to such property in the consolidated corporation, and otherwise carry out the purposes and intent of this agreement.

9. This agreement may be terminated and abandoned at any time prior to the filing of the Certificate of Consolidation if (*a*) the Boards of Directors of both the Constituent Societies agree to such termination; or (*b*) in the judgment of the Board of Directors of either of the Constituent Societies any material litigation shall be pending or threatened against or affecting either of the Constituent Societies which renders it inadvisable to proceed with the consolidation; or (*c*) if this Agreement for Consolidation is not duly approved by the members of the Constituent Societies on or prior to December 31, 1958; or (*d*) if the Board of Directors of either of the Constituent Societies shall resolve that it is inadvisable to comply with any modifications of this agreement required by the court to which the application for an order approving the consolidation shall be filed. In the event of any such termination and abandonment of this agreement, except for the provisions of Paragraph 10 hereof, this agreement shall be void and of no further force and effect, and neither of the Constituent Societies, nor any of their respective directors, officers, members, agents, or employees, shall have any further liability therefor.

10. Upon the filing of the Certificate of Consolidation, or as soon thereafter as may be practicable, the consolidated corporation shall pay all fees and expenses incident to the preparation of this agreement, the carrying of the same into effect, and accomplishing the consolidation provided for herein. In the event that the consolidation shall not become effective for any reason, each of the Constituent Societies shall pay the fees and expenses of their respective counsel, but all other expenses incidental to the preparation for the consolidation shall be divided between and equally borne by the Constituent Societies.

IN WITNESS WHEREOF, AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., *and The American Society of Refrigerating Engineers* have caused this agreement to be signed in their respective corporate names and to be subscribed by their respective presidents, and their respective corporate seals to be hereunto affixed and attested by their respective secretaries.

AMERICAN SOCIETY OF HEATING AND
AIR-CONDITIONING ENGINEERS, INC.,

By ELMER R. QUEER
President

Attest:
AUBRY V. HUTCHINSON
Secretary

The American Society of Refrigerating Engineers,

By CECIL BOLING
President

Attest:
ROBERT C. CROSS
Secretary

STATE OF PENNSYLVANIA:
SS:
COUNTY OF CENTRE:

On this 31st day of October, 1958, before me came ELMER R. QUEER, who, being by me duly sworn, did depose and say that he resides at University Park, Pennsylvania; that he is the President of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., the corporation described in and which executed the foregoing agreement; that he knows the seal of said corporation; that the seal affixed to said instrument is such corporate seal; that it was so affixed by order of the board of directors of said corporation; and that he signed his name thereto by like order.

ELMER R. QUEER (*signed*)

Mary Lou Baney
Notary Public

STATE OF CONNECTICUT:
SS:
COUNTY OF HARTFORD:

On this 3 day of November, 1958, before me came CECIL BOLING, to me known, who, being by me duly sworn, did depose and say that he resides at West Hartford, Conn.; that he is the President of *The American Society of Refrigerating Engineers*, the corporation described in and which executed the foregoing agreement; that he knows the seal of said corporation; that the seal affixed to said instrument is such corporate seal; that it was so affixed by order of the board of directors of said corporation; and that he signed his name thereto by like order.

CECIL BOLING (*signed*)

Lee G. Smith
Notary Public

EXHIBIT A

BY-LAWS FOR PROPOSED CONSOLIDATED SOCIETY AMERICAN SOCIETY OF HEATING, REFRIGERATING, AND AIR-CONDITIONING ENGINEERS, INC.

ARTICLE I—Incorporation

Section 1.1 This is a corporation duly organized and existing under and pursuant to the Membership Corporations Law of the State of New York, on (date), and is hereafter called the "Society."

Section 1.2 The object of the Society is to advance the arts and sciences of Heating, Refrigeration, Air Conditioning, and Ventilation, and the allied arts and sciences, for the benefit of the general public.

ARTICLE II—Government

Section 2.1 The Society shall be governed by the Laws of the State of New York, its Certificate of Consolidation, its By-laws, the Rules promulgated by the Board of Directors in harmony therewith, and all amendments to the foregoing.

Section 2.2 The Society shall neither approve any engineering project or commercial product, nor allow its imprint or name to be used in any commercial work or business, except that it shall be permissible for a manufacturer to state in any manner deemed proper that a product has been tested, or tested and rated, in accordance with an ASHRAE Standard, giving the number of the Standard, however, the Society shall not engage in the testing or rating of such products in behalf of any manufacturer.

Section 2.3 Matters pertaining to politics, religion, or solely to trade shall not be discussed at any meeting of the Society, nor be included in any of its publications.

Section 2.4 As used in these By-laws, the use of "Member" (with a capital M) refers to a member of that grade or higher as set forth in *Section* 3.1, and the use of "member" (with a small m) refers to a member of any grade.

Section 2.5 Official stationery of the Society shall be used only for its official business, and only by its officers, directors, committee members, Chapter and Branch officers, Chapter and Branch committee members, and members of its staff.

Section 2.6 The administrative year of the Society shall begin at the Annual Meeting upon the installation of the new officers and terminate at the following Annual Meeting upon the installation of their successors. The fiscal year shall be determined by the Board of Directors.

Section 2.7 All business meetings of the Society, Board of Directors, Executive and other Committees shall be governed by the rules of procedure contained in Roberts Rules of Order, Revised, where the same are not inconsistent with the law or the provisions of the Certificate of Consolidation, the By-laws, or special rules of order of the Board of Directors.

Section 2.8 The Board of Directors shall designate one or more of the publications of the Society for the publishing of official notices to the members.

ARTICLE III—Membership

Section 3.1 The grades of membership in the Society shall be designated as follows:

(*A*) Honorary Members, (*B*) Presidential Members, (*C*) Life Members, (*D*) Fellows, (*E*) Members, (*F*) Associate Members, (*G*) Affiliates, and (*H*) Students.

Section 3.2 Honorary Member—Any notable person of preeminent professional distinction may be elected an Honorary Member.

Section 3.3 Presidential Member—Upon the installation of his successor, the outgoing President of the Society shall become a Presidential Member. Past Presidents of either predecessor society shall become Presidential Members.

Section 3.4 Life Member—A member who has rendered distinguished service to the Society; has been a Member in good standing for thirty (30) years; and who has attained the age of sixty-five (65) years. He shall retain all the rights and privileges of his former membership grade. He shall not be required to pay any annual dues or any other fees.

Section 3.5 Fellow—A Member who has attained unusual distinction in the arts relating to the sciences of heating, refrigeration, air conditioning or ventilation, or the allied arts and sciences, or in the teaching of major courses in said arts and sciences, or who by reason of invention, research, original work, or as an engineering executive on projects of unusual or important scope, has made substantial contribution to said arts and sciences, and who has attained the age of forty-five (45) years, and has been in good standing as a Member for a period of at least ten (10) years prior to the date of his proposal for Fellow grade.

Section 3.6 A Member, at the time of his admission or advancement;

(*A*) Shall be a graduate of an engineering curriculum accredited by the Engineers Council for Professional Development and approved by the Board of Directors, or approved by the Board of Directors, and shall have had no less than six (6) years active practice in the professions of engineering or teaching, or both, of which five (5) years shall have been in responsible charge of such teaching or engineering work, and who is qualified to direct such work or carry on important research or design in the engineering field; or,

(*B*) If not such a graduate, shall have equivalent attainments including at least twelve (12) years active practice in the professions of engineering or teaching, or both, of which five (5) years shall be in responsible charge of such teaching or engineering work, all of a character satisfactory to the Board of Directors.

A license to practice Professional Engineering issued by a legally authorized body whose requirements as to education and active practice are considered satisfactory and adequate by the Board of Directors may be considered equivalent to fifty (50) percent of the active practice requirements.

Section 3.7 An Associate Member shall at the time of his admission or advancement to the grade, be a graduate of an engineering curriculum accredited by the Engineers

Council for Professional Development and approved by the Board of Directors, or approved by the Board of Directors. If not such a graduate, he shall have equivalent attainments including at least eight (8) years of engineering experience, all of a character satisfactory to the Board of Directors.

Section 3.8 An Affiliate shall have had experience in technical matters, design, operation, or maintenance in heating, refrigerating, air conditioning or ventilating fields, or shall have an interest in the advancement of the Society's aims, and shall possess sufficient qualifications to cooperate with heating, refrigerating, air conditioning or ventilating engineers in the advancement of the knowledge relating to heating, refrigerating, air conditioning, or ventilating engineering and their application.

Section 3.9 A Student shall be a person matriculated in a degree-granting or graduate school with a curriculum accredited by the Engineers Council for Professional Development or the Engineering Institute of Canada, or in a school with a curriculum recommended by the Education Committee and approved by the Board of Directors and pursuing a course of study in preparation for the engineering profession. The Student status shall terminate one year after graduation from the school, or one year from the time he leaves the school.

Section 3.10 The voting membership shall consist of Honorary Members, Presidential Members, Life Members, Fellows, Members, and Associate Members.

All members thus entitled to vote may be called herein "voting members."

Affiliates and Students shall have no right to vote.

Section 3.11 No member shall describe himself in connection with the Society in any advertisement, letterhead, printed matter, or any other manner other than as an Honorary Member, Presidential Member, Life Member, Fellow, Member, Associate Member, Affiliate, or Student, as the case may be, except in official business of the Society.

Section 3.12 The rights and privileges of a member shall be personal to himself and shall not be delegated or transferred, except that each member entitled to vote may vote in person or by written proxy given to another member entitled to vote and dated within three months, which proxy shall be subject to the provisions as set forth in *Section* 6.1.

Section 3.13 All right, title and interest of a member in the Society, or its property, shall cease on the termination of his membership by death, resignation or otherwise, and shall vest in the Society.

Section 3.14 Each member, upon his election to membership, shall thereupon be bound by the provisions of the Certificate of Consolidation, by-laws, and rules of the Board of Directors, and all amendments thereto.

Section 3.15 The emblem of membership in the Society shall be worn only by members in good standing.

Section 3.16 Any member may resign at any time by his written request received by the Executive Secretary, provided his dues are paid in full.

Section 3.17 If any Fellow, Member, Associate Member, or Affiliate shall fail to pay his dues by October 1* he shall be classed as delinquent, and if a voting member shall lose his right to vote; if such dues are not paid by January 1* he shall be classed as not in good standing and his membership shall be suspended; if such dues are not paid by June 1* the Executive Secretary shall notify the suspended member by registered mail that unless such dues are paid by June 30* he shall cease to be a member of the Society, and upon his failure to cure such default by June 30* his membership in the Society shall cease; if any Student shall fail to pay his dues by June 30* the delinquent Student's

* These dates in *Section* 3.17 were fixed at the Executive Committee Organization Meeting held on January 29, 1939.

membership shall cease and the Executive Secretary shall notify such Student by registered mail that his membership in the Society has ceased; provided that upon written application satisfactorily explaining a default, accompanied by payment of dues, the Board of Directors may, in its discretion, rescind any forfeiture of membership.

Section 3.18 A former member who has resigned, or who has been dropped from membership, may be reinstated by payment of the same fees charged a new member, or may be reinstated as of his original date of membership if he pays all dues which would have accrued.

Section 3.19 The Board of Directors may, by a two-thirds vote of all the members thereof, censure, suspend or expel any member for misconduct in his relations to the Society, after written preferment of charges, thirty (30) days' written notice of hearing sent by registered mail, and an adequate opportunity to be heard before the Board of Directors or a committee of one or more Members designated by the Board of Directors.

Section 3.20 All applicants for admission to the Society, or for advancement in grade of membership except such as are conferred as an honor, shall make application in such form and with such information as shall be required by the Board of Directors.

Section 3.21 Membership in the Society and advancement in membership grade shall be by vote of the Board of Directors on proposals, or applications, or as set forth below. An unanimous vote by secret ballot shall be required for election to Honorary Member grade.

Section 3.22 All applications for admission to the Society, or advancement in grade, shall be referred to the Admissions Committee for investigation and report to the Board of Directors with recommendation as to grade. As soon as practicable after the report, the Board of Directors shall act upon each application by letter ballot. Two (2) disapprovals shall reject any applicant. One (1) disapproval shall require the resubmission of the application to the Admissions Committee for further study and it shall make recommendation to the Board of Directors and the Board of Directors shall vote upon the application at its next regular meeting.

Section 3.23 A person whose Student grade has terminated may be advanced to Associate Member or Affiliate grade by the Board of Directors upon application providing proof of his qualifications therefor.

Section 3.24 The grade of Honorary Member shall be conferred on no more than three (3) persons in any calendar year. The grade of Fellow shall be conferred on no more than fifteen (15) Members in any calendar year.

Section 3.25 Before submission to the Admissions Committee of an application for election, or advancement to Affiliate, Associate Member, or Member, the name of the applicant shall be published in an issue of the Society's official journal.

Section 3.26 Nomination for Fellow or Life Member, which are honors conferred by the Society, shall be proposed by recommendation of the Awards Committee to the Board of Directors, or by petition of not less than ten (10) Fellows and Members to the Board of Directors. Election shall be by the Board of Directors and more than two (2) negative votes shall defeat the proposal.

ARTICLE IV—Fees and Annual Dues

Section 4.1 Initiation and advancement fees and annual dues shall be fixed by the Board of Directors from time to time and shall be payable as determined by the Board of Directors, and shall be published periodically in the official publication of the Society.

Section 4.2 The Board of Directors may provide for a paid-up membership by one payment of a sum fixed by the Board of Directors and such member shall retain for life

all the rights and privileges of his membership grade before such payment was made, unless otherwise deprived thereof.

Section 4.3 Any member who has paid dues for thirty (30) years shall be permanently exempt from the payment of further dues, and shall retain all the rights and privileges of his membership grade, unless otherwise deprived thereof.

Section 4.4 The Board of Directors may waive the dues of any member for good and sufficient reasons.

Section 4.5 Members of all grades shall receive the official journal of the Society, Members of all grades, except Student and dues paying members whose pro-rated dues amount to less than fifty percent (50%) of a Member's dues, shall be entitled to receive the Society's other publications as authorized by the Board of Directors.

ARTICLE V—The Board of Directors and Officers

Section 5.1 The Board of Directors shall consist of the President, the 1st Vice President, the 2nd Vice President, the Treasurer, the immediate Past President; nine (9) Directors-at-Large, with equitable distribution from the three (3) major areas of membership interest, namely, (1) heating, (2) refrigeration, and (3) air conditioning and ventilation; and ten (10) Regional Directors, each from his respective Region. The nine (9) Directors-at-Large shall be divided into three (3) groups of three (3) each in each area of membership interest, and the members in each group shall hold office for three (3) years, and until their successors shall have been elected and installed. Three (3) (3) Directors-at-Large shall be elected at each Annual Meeting, and also such additional number, if any, as may be necessary to fill vacancies.

The ten (10) Regional Directors shall be elected for a staggered term of three (3) years; a 3-3-4 sequence of the group of ten (10) to be elected annually at the Annual Meeting.

Section 5.2 The Board of Directors shall hold regular meetings at approximately the time of the Semi-Annual and Annual Meetings of the Society.

Special Meetings of the Board of Directors may be called by the President or by three (3) members of said Board. The Board of Directors shall keep a record of its proceedings and shall report on its activities at each meeting of the Society and at the Annual Meeting shall present a written report as required by the Membership Corporations Law of the State of New York.

A quorum of the Board of Directors shall consist of thirteen (13) of its members.

Section 5.3 The Board of Directors shall have full and complete management and control of the activities, properties, and funds of the Society, subject to the provisions of law, the Certificate of Consolidation, and the By-laws.

The Board of Directors may, in its discretion, refer to the Society any important question pertaining to the Society, and shall refer any such questions to the Society upon a majority vote taken at a stated or Special Meeting of the Society.

Section 5.4 The Officers of the Society shall be President, 1st Vice President, 2nd Vice President, Treasurer, and Executive Secretary. The President shall not be eligible for immediate re-election to that office. Elected officers shall receive no salary, emolument or compensation for services rendered to the Society, and shall serve for one (1) Society year or until their respective successors shall be elected and installed.

Section 5.5 All officers of the Society shall perform the duties customarily attaching to their respective offices under the laws of the State of New York, and such other duties and services incident to their respective offices as are delegated to them in these By-laws and as may from time to time be assigned to them by the Board of Directors.

Section 5.6 The terms of all Officers and Directors shall commence upon their installation during the Annual Meeting and shall continue until their successors have been elected and installed.

Section 5.7 Only Honorary Members, Presidential Members, Life Members, Fellows and Members shall be eligible for election as President, Vice President, Treasurer, or Director.

Section 5.8 At all meetings of the Society and of the Board of Directors, the President, or in his absence the Vice Presidents in order of seniority, or in their absence the Treasurer, or a Member selected by the Board of Directors, shall preside.

A vacancy in the office of President shall be filled by the ranking Vice President, or in the event of vacancies in the offices of both Vice Presidents, by election by the Board of Directors. In the event of a vacancy in any other office or directorship the same shall be filled by election by the Board of Directors until the next Annual Meeting.

Section 5.9 The Treasurer shall have custody of the funds of the Society and the Society's books of account, which shall be opened to the inspection of any member of the Board of Directors.

Section 5.10 The expenditures of the Society's funds shall be governed by the Budget as approved, modified, or from time to time amended, by the Board of Directors, and no additional expenditures shall be made without the approval of the Board of Directors.

Section 5.11 The Treasurer, the Executive Secretary, and all Officers, Agents or Employees authorized by the Board of Directors to endorse or execute drafts for the payment of money, shall give a bond in a penal sum and with sureties approved by the Board of Directors, for the faithful performance of his or their duties, the premiums therefor, to be paid by the Society.

Section 5.12 The Executive Secretary shall be appointed by the Board of Directors under an employment agreement approved by the Board of Directors, fixing his salary, term of employment, and other conditions. The Executive Secretary shall be subject to removal by a vote of two-thirds (⅔) of the Board of Directors present and voting by secret ballot as a meeting.

Section 5.13 The Executive Secretary shall act as secretary of the Board of Directors and of the Executive Committee and as an ex-officio member of all other committees, except the Nominating Committee. He may take part in the deliberations of all these bodies; but shall not have a vote therein. The Executive Secretary shall, under the supervision of the Finance Committee, have charge of the collections and of keeping the books. He shall present, at the meeting of the Society following the close of the fiscal year a summary of membership enrollment and other pertinent records, and shall perform such other duties as may be assigned to him by the Board of Directors, the Executive Committee, or the President.

Section 5.14 After the close of the fiscal year, the accounts of the Society shall be audited by a certified public accountant approved by the Board of Directors, and the auditor's report shall be presented by the Treasurer at the Annual Meeting of the Society, and shall be published in the official publication.

Section 5.15 Unless waived in writing or by telegraph or by cable, notice of any regular or special meeting of the Board of Directors shall be given in writing, mailed to the last known address of each member of the Board of Directors, by the Executive Secretary or the President, or the three (3) members of the Board of Directors calling the meeting, not less than fifteen (15) nor more than thirty (30) days before the date fixed for the meeting.

ARTICLE VI—Elections

Section 6.1 Voting at any meeting may be in person or by proxy, but only voting members in good standing of the Society shall be eligible to act as proxies. A proxy shall not be valid after more than three (3) months from its date of execution. Voting for election of officers, Board of Directors members, on proposals to amend these By-laws, and on questions referred to the Society pursuant to Article V, *Section* 5.3 shall be by secret ballot. In the event of any tie vote, the Board of Directors shall decide the vote.

Section 6.2 Together with notice of the Annual Meeting, the Executive Secretary shall forward appropriate proxies and ballots to members entitled to vote. The proxies and ballots shall contain spaces for write-in names.

Section 6.3 The polls for election shall be opened at the opening of the Annual Meeting and shall remain open for a period of five (5) hours. Thereafter the ballots shall be opened by three (3) inspectors of election appointed by the President, who shall be authorized to fill any vacancy occurring among such inspectors. The inspectors of election shall consider ballots and votes to be valid provided the intent of the voter is clear. The result of the vote shall be reported by the inspectors of election in writing, and shall be announced by the President on the second day of the Annual Meeting, whereupon the terms of the inspectors of election shall expire.

The elected candidates shall be installed during the Annual Meeting.

Section 6.4 There shall be published in the official publication of the Society, three (3) months prior to the Annual Meeting, the names and qualifications of the consenting nominees as submitted by the Nominating Committee; and two (2) months prior to the Annual Meeting, the names and qualifications of the consenting nominees submitted in proper time by any petitioning group.

Section 6.5 The President, Vice Presidents, and the Treasurer shall each be elected to serve in their respective offices for a term of one (1) year. The Directors shall each be elected for a term of three (3) years, one-third (⅓) of them to be elected each year, or as specified in Article V, *Section* 5.1.

ARTICLE VII—Meetings

Section 7.1 The Annual Meeting of the Society shall commence on a day and at a time fixed by the Board of Directors, at approximately the middle of the calendar year, and shall continue from day to day until adjourned.

Semi-Annual Meetings shall be held at such times as may be fixed by the Board of Directors. Special Meetings may be called at any time by the Board of Directors, and shall be called by the Board of Directors upon the written request of the President, or of fifty (50) members of the Society. Meetings shall be held at such place or places as the Board of Directors may designate, and shall be governed by the laws of the State of New York, the Certificate of Consolidation, and these By-laws. At any meeting of the Society, the presence of fifty (50) members entitled to vote in person or by proxy shall constitute a quorum.

Section 7.2 Notice of the Annual and Semi-Annual Meetings of the Society shall be published in the official publication of the Society. Notice of any Special Meeting of the Society shall be given in writing by the Executive Secretary and mailed, postage prepaid, not less than twenty (20) nor more than forty (40) days before the date fixed for the meeting, to each member of the Society at his last known address appearing on the records of the Society.

Notice of a Special Meeting shall state the purpose for which the meeting is called, and no business other than that set forth in the notice shall be entertained or transacted thereat.

ARTICLE VIII—Committees

Section 8.1 Committees of the Society shall consist of General Committees and Special Committees.

Section 8.2 Unless otherwise provided, the General Committees, and the respective Chairmen thereof, shall be designated by the President with the approval of the Board of Directors as soon as practicable after the close of the Annual Meeting of the Society.

Section 8.3 The Board of Directors shall prescribe the qualifications of, and the number of committees, unless otherwise provided.

Section 8.4 The Board of Directors may from time to time create other committees of one (1) or more members, and define their powers and duties, and it may abolish any such committees.

Section 8.5 The Chairman and Vice Chairman of each General or Special Committee shall be a Member or Associate Member.

Section 8.6 The President may appoint any person or persons to serve in a consulting capacity to any General or Special Committee.

Section 8.7 The names of the General Committees of the Society shall be as follows:

Executive
Finance
General and Administrative
Coordinating
Technical Coordinating
Divisional Advisory
Regions Central
Chapters Regional
Long Range Planning
Advisory Board—Past Presidents
Admissions and Advancement
Charter and By-laws
Exposition
Education
Honors and Awards
International Relations
Meetings Arrangements
Membership Development
Nominating
Professional Development
Program
Publications
Public Relations
Research and Technical
Standards

Section 8.8 The duties and functions of each of the Committees shall be as follows:

Section 8.8.1 The Executive Committee shall consist of the President, who shall be its Chairman, the immediate Past President, the 1st Vice President, the 2nd Vice President, and the Treasurer. It shall meet at the call of the President, or upon request of any two (2) members of the committee. It shall investigate and make reports and recommendations to the Board of Directors regarding matters relating to the Society or any member, or members thereof. During intervals between Board of Directors meetings the Executive Committee shall exercise the powers of and act for the Board of Directors.

Section 8.8.2 The Finance Committee shall consist of five (5) Members including the Treasurer, one Vice President and three (3) others, each of these three (3) to serve a term of three (3) years. The President shall appoint one (1) of these last three (3) committeemen each year, shall select the Vice President who is to serve, and shall fill any vacancies. One of these presidential appointees shall be a member of the Board of Directors.

The Finance Committee, under the direction of the Board of Directors, shall supervise and control the financial affairs of the Society and its books of account. It shall survey, investigate and analyze all financial requirements and expenditures, scrutinize all Budget estimates, and prepare the Budget for submission to the Board of Directors.

The Finance Committee shall submit to the Board of Directors prior to the end of the fiscal year a Budget of estimated income and expenditures of the Society and all the committees thereof, for the succeeding fiscal year.

Section 8.8.3 The General and Administrative Coordinating Committee shall consist of the Chairmen and Vice Chairmen of the following committees:

Admissions and Advancement, Education, Honors and Awards, Meetings Arrangements, Membership Development, Professional Development, and Public Relations; and shall have a Vice President as Chairman. It shall be its duty to promote and coordinate participation by the members in the activities of the Society falling within the purview of said committees.

Section 8.8.4 The Technical Coordinating Committee shall consist of the Chairmen and Vice Chairmen of the Program, Publications, Research and Technical, and Standards Committees, and shall have a Vice President as Chairman. In addition, there shall be three (3) Members-at-Large appointed by the President of the Society. It shall be the duty of the Committee to coordinate the activities of the committees named in this Section.

Section 8.8.5 The Divisional Advisory Committee shall consist of the nine (9) Members-at-Large of the Board of Directors, three (3) each being elected from the (1) heating, (2) refrigeration, (3) air conditioning and ventilation areas of membership professional interest, and each of whose major interests shall be in the particular category represented. It shall be the duty of the Committee to review, and advise the Board of Directors, when it deems that any area of membership professional interest is not being duly represented or recognized.

It is the duty of this Committee to report in detail at each meeting of the Board of Directors the degree to which each area of professional interest is being served or recognized in the activities of the Society.

Wherever twenty-five (25) or more members in a Chapter, or two hundred (200) or more members nationally signify their desire to carry out a program at Chapter level, or national level, respectively, in a specialty within the fields defined in *Section* 1.2, this Committee and the Board of Directors shall provide all reasonable facilities customarily provided to other specialties.

Section 8.8.6 The Regions Central Committee shall consist of the 2nd Vice President, as its Chairman, and the Regional Directors. The said committee shall consider and report to the Board of Directors on the activities of Chapters and make recommendations to the Board of Directors concerning the policies, procedures and operation of the Society and its Chapters; it shall coordinate the activities of the Chapters Regional Committees; and it shall investigate applications for the creation of Chapters, and report thereon to the Board of Directors.

Section 8.8.7 The Advisory Board, consisting of the Past Presidents shall advise the Society on matters of basic Society policy or interest.

Section 8.8.8 Chapters Regional Committees, each serving one Regional Area, and each consisting of the Regional Director for the area and one (1) member and one (1) alternate member selected by each Chapter therein, to serve for a term of one (1) year. The said committees shall solicit from the Chapters and Branches within their respective Regional Areas recommendations concerning the policies, procedures, and operation of the Society, its Chapters and Branches, review the same, and make recommendations thereon to the Regions Central Committee. Said committees shall select the Members and alternates to serve on the Nominating Committee, and duly notify the Executive Secretary of such selections. The alternate members of Chapters Regional Committees may be present at committee meetings and participate in the deliberations thereof, but shall not vote therein except in the absence of the committee members for whom they respectively are alternates. The said committees shall hold committee meetings not earlier than forty-five (45) days after the Annual Meeting. Each Regional Director shall be the Chairman of the Committee serving his Regional Area.

Section 8.8.9 The Long-Range Planning Committee shall consistently make the necessary studies to prepare for and recommend to the Board of Directors, long-range

planning on the aims and activities of the Society which in the opinion of the Committee would affect the future welfare and growth of the Society.

Section 8.8.10 The Admissions and Advancement Committee shall receive all applications for membership and advancement and recommend the names of all applicants and the grade for which it considers them to be eligible, except membership grades conferred as an honor, and report to the Board of Directors. The correspondence, information obtained, and proceedings of said committee shall be secret and confidential, and its records concerning unsuccessful applicants shall be destroyed within a reasonable time.

Section 8.8.11 Charter and By-law Committee, consisting of three (3) Members, which shall consider all matters requiring possible changes in the Certificate of Consolidation, By-laws, Rules and Regulations, and make recommendations thereon to the Board of Directors.

Section 8.8.12 The Education Committee shall advise the Society in its educational programs, in its relations with educational institutions, and those matters pertaining to the requirements for membership of students and graduates therefrom, and related matters.

Section 8.8.13 The Exposition Committee shall coordinate the activities of the Society with expositions, shows, and the like, related to the arts of heating, refrigeration, air conditioning and ventilation.

Section 8.8.14 The Honors and Awards Committee shall recommend to the Board of Directors the candidates for all awards for contributions to the sciences and arts of heating, refrigeration, air conditioning and ventilation, or closely allied fields, and for articles appearing in the official publication, and other gifts or awards including membership grades conferred as an honor. The committee shall include a Past President.

Section 8.8.15 The International Relations Committee shall represent the Society in its cooperation and relationship with foreign engineering societies and groups and make recommendations thereon to the Board of Directors.

Section 8.8.16 The Meetings Arrangements Committee shall study the suitability of locations for meetings of the Society and determine that proper facilities are available, and shall make its recommendations to the Board of Directors.

Section 8.8.17 The Membership Development Committee shall publicize the aims, activities, achievements and scientific and educational purposes of the Society toward the end that persons duly qualified shall apply for membership therein.

Section 8.8.18 The Nominating Committee shall consist of eighteen (18) members; as follows: One (1) member with one (1) alternate from each Region of the Society selected by the Chapters Regional Committees; six (6) members with six (6) alternates selected by the Board of Directors of whom two (2) members shall represent each of the three (3) major areas of membership interest, namely, (1) heating, (2) refrigeration, (3) air conditioning and ventilation; one (1) member selected by the President from the last prior Nominating Committee, and the last preceding Past President who is not a member of the Board of Directors, who shall be its Chairman and shall have the right to vote.

Nominations of officers and members of the Board of Directors, other than those nominated by the Nominating Committee, may be made in writing by not less than fifty (50) members eligible to vote, upon presentation of such nominations, with each nominee's consent, to the Executive Secretary at least sixty (60) days prior to the first session of the Annual Meeting, whereupon the nominees' names shall be placed upon the ballot with a notation that they are presented by members independent of the Nominating Committee.

Section 8.8.19 The Professional Development Committee shall promote the professional development of the members amid the profession of engineering in general.

Without engaging in propaganda or otherwise influencing or intending to influence legislation, this Committee shall endeavor to obtain greater public recognition of the profession of engineering in the advancement of human welfare.

Section 8.8.20 The Program Committee shall plan the general character of all technical meetings of the Society, and shall solicit, receive, and select papers for presentation at such meetings.

Section 8.8.21 The Publications Committee, subject to the direction of the Board of Directors, shall formulate the editorial policies of the Society and all of its publications. The chairman may appoint subcommittees of one (1) or more members to review and report to the committee on the quality and appropriateness for publication of papers and bulletins intended for presentation or presented at Society meetings and the discussions thereof. In the performance of its functions the said committee and its subcommittees shall be subject to the following conditions: (*a*) that the data recommended for publication shall tend toward the professional education of the individual engineer; (*b*) that such data shall be free from commercial bias; and (*c*) that such data shall tend to advance for the public benefit the sciences relating to the arts of heating, refrigeration, and air conditioning and ventilation, or the allied arts and sciences.

Section 8.8.22 The Public Relations Committee shall publicize the aims, activities and achievements of the Society and its scientific and educational purposes with the object of cultivating and stimulating public and members' interest in the Society and its affairs.

Section 8.8.23 The Research and Technical Committee, subject to the direction of the Board of Directors, shall conduct and coordinate basic research and technical studies in the fields of heating, refrigeration, and air conditioning and ventilation, subject to the proviso that these activities shall be devoted to the public welfare and general benefit, and shall not be designed to promote any individual, private, or commercial interests.

In addition to the research activities, this committee shall plan for and have charge of the activities of the technical committees appointed to further the advancement of the arts and sciences of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences for the public benefit. It shall determine the scope of the activities of each of these technical committees.

The committee shall consist of twelve (12) members nominated by the Board of Directors, and elected by vote of the Board of Directors. Four (4) members shall be elected each year to serve for a term of three (3) years.

The Chairman, on the recommendation of the Committee, shall appoint such Technical Committees as may be deemed expedient to carry out the objectives of the Committee, or to advise it on specific projects.

Section 8.8.24 The Standards Committee shall have charge of the selection, development, preparation, and submittal to the Board of Directors and the membership, of all codes and standards in the fields of heating, refrigeration, and air conditioning and ventilating engineering, and all revisions thereof, to be considered for adoption by the members. It shall cooperate with other organizations in the development, preparation, and adoption of codes and standards. Adoption of a code, or a standard, or a revision thereof, shall require the recommendation of the Board of Directors and approval by the members at a regular meeting. Notice that each proposed code or standard or revision thereof to be considered for adoption shall be given in writing, or by printing in the official publication, to each member with a copy of the same at least thirty (30) days in advance of the meeting at which its adoption will be considered.

The activities of the Committee shall be solely for the development of engineering science, and the Committee shall not engage in influencing enactment of building or other codes, or in propaganda, or other activities designed to influence legislation.

Section 8.9 All General Committees, except the Nominating Committee shall render to the Board of Directors, prior to the Annual Meeting of the Society, annual reports of

their activities and shall submit progress reports at other times on request of the President.

Section 8.10 Any Committee may be permitted by the President, with the approval of the Board of Directors, to coordinate its activities with other organizations or groups having interests kindred to those of the Society.

Section 8.11 Each General Committee at the request of the Board of Directors shall prepare an Operational Procedure which shall govern its activities after approval by the Board of Directors. These may be amended as required, with the approval of the Board of Directors.

Section 8.12 The Board of Directors may by a two-thirds (⅔) vote remove a member of any committee excepting Regions Central and Nominating Committees.

Section 8.13 Each committee's actions, proceedings, findings, conclusions and reports shall be subject to the direction and review of the Board of Directors, and the Board of Directors may take such steps, or see that such steps are taken by the committees as may be appropriate to comply with the Charter and By-laws, and to make effective any resolution adopted by the Society or any resolution, rule or directive of the Board of Directors.

Section 8.14 If any doubt or controversy should arise as to whether a particular subject or matter is within the jurisdiction of a committee or whether any action should be taken by a committee, or in the case of a committee tie vote, the same shall be settled and determined by the Board of Directors.

ARTICLE IX—Chapters and Regions

Section 9.1 The Board of Directors may establish Chapters which shall operate under the provisions of the Charter and the By-laws of the Society, and the Rules and Regulations of the Board of Directors.

Section 9.2 Branches of the Society may be established, operated, and maintained under the direction and in the discretion of the Board of Directors. Such Branches shall be Special Branches, consisting of groups within continental North America; Overseas Branches, consisting of groups outside of continental North America, and Student Branches.

Section 9.3 Chapters of the Society shall be grouped in geographical areas by the Board of Directors and each such area shall be designated as a Region. The number and delineations of Regions and changes therein shall be published in the official publication.

Section 9.4 A Chapter or Branch of the Society may be authorized upon approval by the Board of Directors of a written petition of twenty-five (25) members of the Society and the adoption of By-laws, based on the Model By-laws for Chapters or Branches, which have been approved by the Board of Directors.

Only members of the Society in good standing shall be eligible to become and remain Chapter or Branch members. Chapter or Branch members shall hold the same grade of membership in the Chapter or Branch as are held by them in the Society. No member shall vote or hold office concurrently in more than one (1) Chapter or Branch of the Society. All grades of Chapter or Branch members, except Students, shall be eligible to vote and hold office in Chapters or Branches.

Section 9.5 A charter shall be granted by the Board of Directors to each duly authorized Chapter or Branch. It shall be signed by the President and attested by the Executive Secretary.

Section 9.6 The elected officers of Chapters or Branches shall receive no salary, emolument or compensation for their services as such. Chapters or Branches shall not act for the Society or subject the Society to any financial or other obligation, except such as the Society or the Board of Directors may by resolution specifically assume. Notice to the foregoing effect shall be imprinted on the stationery used by each of the Chapters or Branches. Each Chapter or Branch shall promptly file a copy of its Minutes with the Executive Secretary of the Society and make report to said Secretary of all of its proceedings. Each Chapter and Branch shall file with the Chairman of its respective Chapters Regional Committee its recommendations concerning the policies, procedures and operation of the Society, its Chapters and Branches. No contributions, except dues, shall be received or solicited by Chapters or Branches without the written approval of the Board of Directors. Chapters or Branches shall not issue publications or use the Society's name or emblem or Chapter or Branch insignia, without the approval of the Board of Directors. Chapters or Branches shall give no recommendations, endorsements or approvals of any scientific, literary, mechanical or engineering product for the promotion of private interests.

Section 9.7 Each Chapter or Branch shall annually elect and install officers prior to the Annual Meeting of the Society. Each Chapter may be provided with funds by the Society as the Board of Directors may deem appropriate.

Section 9.8 The charters of Chapters, Special Branches and Student Branches, and Overseas Branches, may be revoked by a two-thirds (⅔) vote of all the members of the Board of Directors after written preferment of charges, sixty (60) days written notice of hearing sent by registered mail to the President of the Chapter or Branch, and an adequate opportunity to be heard before the Board of Directors or a committee of three (3) or more Members designated by the Board of Directors.

ARTICLE X—Funds

Section 10.1 Society Reserve Fund. Admission fees and such other funds as may from time to time be recommended by the Finance Committee and allocated by the Board of Directors shall be set aside and the principal thereof maintained as a Society Reserve Fund. The Board of Directors is authorized and empowered, in any fiscal year in which the Society's revenues may be insufficient to meet expenses, to utilize a maximum of twenty percent (20%) of the Society Reserve Fund as valued on the first day of the fiscal year in which such a withdrawal may be required.

Section 10.2 Allocation of Dues for Research. Unless changed by the Society at an Annual or Special Meeting, a percentage determined by the Board of Directors of the dues shall be allocated for basic or fundamental research in the principles and laws underlying matters in the arts relating to the sciences of heating, refrigerating, and air conditioning and ventilation, and the allied arts and sciences.

Section 10.3 Investment of all funds of the Society shall be made by the Treasurer upon direction of the Finance Committee or the Board of Directors, or under such other arrangements as are approved by the Board of Directors.

ARTICLE XI—Amendments

Section 11.1 Prerequisites. These By-laws may be amended by a two-thirds (⅔) vote of the Society at an Annual or Semi-Annual Meeting thereof, provided that written notice of the proposed amendment, subscribed by two-thirds (⅔) of the members of the Board of Directors or by fifty (50) Members, be given at a previous stated or Special Meeting, and that notice thereof as pertinently amended by majority vote at said stated or Special Meeting be also given by the Executive Secretary in the notice of the Annual Meeting.

Section 11.2 Renumbering. The Board of Directors may, by a two-thirds (⅔) vote, renumber existing articles or Sections of these By-laws.

ARTICLE XII—Interim Provisions

Inasmuch as this Society is a corporation formed by the consolidation of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and *The American Society of Refrigerating Engineers*, known as the constituent corporations, and as ASHAE and *ASRE* respectively;
and
inasmuch as the members of the constituent Societies desire to continue the officers and directors of the respective constituent corporations for the welfare and benefit of the consolidated corporation and its members;
and
Notwithstanding any provisions of these By-laws of this consolidated corporation to the contrary, it is understood and agreed that the By-laws of this consolidated corporation shall include the following interim provisions:

(*a*) The officers and directors shall be persons named and for the terms specified in the schedule which is made a part of the agreement for consolidation of the constituent Societies, and which schedule is made a part hereof;

(*b*) Upon the death, or inability or failure to act, of any of said officers or directors, his successor for his unexpired term shall be elected at the next meeting of the Board of Directors and the person so elected shall be from the same constituent Society as his predecessor; and

(*c*) After the expiration of the terms of all officers and directors as provided for in said schedule, then these interim provisions and this Article XII shall become null and void.

EXHIBIT B

OFFICERS AND BOARD OF DIRECTORS†

JANUARY 1959 TO JUNE 1959

PRESIDENT.....Cecil Boling
1ST VICE PRESIDENT.....Arthur J. Hess
2ND VICE PRESIDENT.....Daniel D. Wile
3RD VICE PRESIDENT.....Walter A. Grant
4TH VICE PRESIDENT.....Robert H. Tull
5TH VICE PRESIDENT.....ASHAE
1ST TREASURER.....ASHAE
2ND TREASURER.....Franklyn Y. Carter

***Hermann F. Spoehrer	**ASHAE	*Willis R. Woolrich
***Elmer R. Queer	ASHAE	*Charles L. Hall
***Carlyle M. Ashley	ASHAE	*William F. Wischmeyer
***Leon Buehler, Jr.	John Engalitcheff, Jr.	*Thomas J. Phillips
**William G. Hole	Charles T. Hamilton	*John W. Chandler
**Peter J. Marschall	Everett P. Palmatier	*Russell A. Baker
**Wesley O. Stewart	Robert G. Werden	*ASRE (1)
**Percy N. Vinther	Carl R. Fagerstrom	*James P. McShane
**Frank H. Faust	Ralph A. Gonzalez	*Charles C. E. Harris
Fred Janssen	James W. Snyder	*Thomas G. Crider
**James W. May	John E. Dube	*William E. Fontaine
Gayle B. Priester	Harold M. Hendrickson	*J. Guy Woodroof
**ASHAE	Si J. Williams, Jr.	*George B. Rottman

TOTAL 47

*ASRE Regional Director. **ASHAE Regional Director. (1) Replacement for T. J. Ammel (resigned)
***These individuals plus all elected officers constitute the Executive Committee.

† In this portion of Exhibit B where the letters ASHAE and ASRE occur in the lists of officers and directors, individuals were designated to fill these posts by action of the Executive Committee of the Consolidated Society (acting for the Board of Directors) at the meeting of the Executive Committee held on January 29 1959.

OFFICERS AND BOARD OF DIRECTORS

JUNE 1959 TO JANUARY 1960

PRESIDENT.....Arthur J. Hess
1ST VICE PRESIDENT.....Daniel D. Wile
2ND VICE PRESIDENT.....Walter A. Grant
3RD VICE PRESIDENT.....Robert H. Tull
4TH VICE PRESIDENT.....ASHAE
1ST TREASURER.....Franklyn Y. Carter
2ND TREASURER.....ASHAE

***Cecil Boling	**ASHAE	*Willis R. Woolrich
***Hermann F. Spoehrer	ASHAE	*Charles L. Hall
***Elmer R. Queer	ASHAE	*William F. Wischmeyer
***Carlyle M. Ashley	Carl R. Fagerstrom	*Charles C. E. Harris
**Frank H. Faust	Ralph A. Gonzalez	*Thomas G. Crider
Fred Janssen	James W. Snyder	*William E. Fontaine
**James W. May	John E. Dube	*J. Guy Woodroof
Gayle B. Priester	Harold M. Hendrickson	*Thomas J. Phillips
**ASHAE	Si J. Williams, Jr.	*George B. Rottman

TOTAL 34

OFFICERS AND BOARD OF DIRECTORS

JANUARY 1960 TO JUNE 1960

PRESIDENT.....Daniel D. Wile
1ST VICE PRESIDENT.....Walter A. Grant
2ND VICE PRESIDENT.....Robert H. Tull
3RD VICE PRESIDENT.....ASHAE
TREASURER

***Arthur J. Hess	**ASHAE	Si J. Williams, Jr.
***Cecil Boling	ASHAE	*Willis R. Woolrich
***Hermann F. Spoehrer	ASHAE	*Charles L. Hall
***Elmer R. Queer	Harold M. Hendrickson	*William F. Wischmeyer
**ASHAE		10 new members

TOTAL 28

OFFICERS AND BOARD OF DIRECTORS

JUNE 1960 TO JANUARY 1961

PRESIDENT.....Walter A. Grant
1ST VICE PRESIDENT.....Robert H. Tull
2ND VICE PRESIDENT.....ASHAE
TREASURER

***Daniel D. Wile	***Cecil Boling	10 Members from period
***Arthur J. Hess	***Hermann F. Spoehrer	January 1960 to June 1960
	***Elmer R. Queer	9 New members

TOTAL 28

OFFICERS AND BOARD OF DIRECTORS

JANUARY 1961 TO JUNE 1961

PRESIDENT—Robert H. Tull
1ST VICE PRESIDENT—ASHAE
2ND VICE PRESIDENT
TREASURER

***Walter A. Grant	***Arthur J. Hess	10 Members from period
***Daniel D. Wile	***Cecil Boling	January 1960 to June 1960
		9 Members from period
		June 1960 to January 1961

TOTAL 27

*ASRE Regional Director.
**ASHAE Regional Director
***These individuals plus all elected officers constitute the Executive Committee

OFFICERS AND BOARD OF DIRECTORS

JUNE 1961 TO JUNE 1962

PRESIDENT—ASHAE
1ST VICE PRESIDENT
2ND VICE PRESIDENT
TREASURER

***Robert H. Tull
***Daniel D. Wile
6 Members from period January 1960 to June 1960

4 New members
9 Members from period June 1960 to January 1961
3 New members

TOTAL 28

*** These individuals plus all elected officers constitute the Executive Committee.

II—Petition for Consolidation

SUPREME COURT OF THE STATE OF NEW YORK, COUNTY OF NEW YORK

In the Matter of the Application of

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

and

The American Society of Refrigerating Engineers,
Petitioners,

for an order pursuant to *Section* 52 of the Membership Corporations Law approving their Agreement for Consolidation under the name

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

and authorizing the filing of the Certificate of Consolidation pursuant to *Section* 50 of said Membership Corporations Law.

The joint petition of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., by Brodek & Eisner, its attorneys, and *The American Society of Refrigerating Engineers*, by Stephen B. Vreeland, its attorney, respectfully shows and alleges:

1. Petitioner, AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., has its principal office at 62 Worth Street, New York 13, N. Y. Said petitioner is a membership corporation, duly organized and existing under and by virtue of the laws of the State of New York; that petitioner's certificate of incorporation was duly filed in the office of the Secretary of State on or about January 24, 1895, under the name THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS; that amended certificates of incorporation were filed by petitioner in the office of the Secretary of State on or about May 20, 1914, March 8, 1946, December 7, 1949, and February 1, 1952; that on or about December 8, 1954, a certificate was filed in the office of the Secretary of State changing the name of said petitioner to AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC. A true copy of the provisions of the certificate of incorporation, as amended, of said petitioner is annexed hereto, made part hereof, and marked *Exhibit A*.

2. Petitioner, *The American Society of Refrigerating Engineers*, has its principal office at 234 Fifth Avenue, New York 1, N.Y. Said petitioner is a membership corporation,

duly organized and existing under and by virtue of the laws of the State of New York; that said petitioner's certificate of incorporation was duly filed in the office of the Secretary of State on or about August 30, 1905; that amended certificates of incorporation were filed by said petitioner in the office of the Secretary of State on or about November 26, 1945, November 23, 1955, and December 28, 1956. A true copy of the provisions of the certificate of incorporation, as amended, of said petitioner is annexed hereto, made part hereof, and marked *Exhibit B*.

3. As shown by said *Exhibits A* and *B* the objects and purposes of petitioners are virtually identical, the purposes of petitioner AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., being to advance the arts and sciences of heating, ventilating, cooling, and air conditioning, and the allied arts and sciences, for the benefit of the general public, and the purposes of petitioner *The American Society of Refrigerating Engineers* being to advance the arts and sciences of air conditioning, heating, refrigeration, and ventilating, and the allied arts and sciences, for the benefit of the general public.

4. Your petitioners are desirous of consolidating under the new name AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC., and petitioners have been advised by their respective counsel, and they verily believe, that the consolidation herein prayed for is authorized by *Section* 50 of the Membership Corporations Law of this State.

5. Your petitioners hereby allege, and verily believe, that the interests of the petitioning corporations, the members thereof, and the public will be promoted and advanced and not adversely affected by the consolidation of the petitioning corporations, and that such consolidation will result in economies in administration and operation and in the expansion and improvement of the services rendered by petitioners to the members and the general public.

6. An Agreement for Consolidation was made and entered into by and between your petitioners as of October 18, 1948. A true copy of said Agreement for Consolidation is annexed hereto, made part hereof, and marked *Exhibit C*. The proposed consolidation and the aforesaid Agreement for Consolidation have been duly approved by your petitioners by the affirmative votes of over two-thirds of the members of each of your petitioners (to-wit, 93% and 73%, respectively) entitled to vote, present in person or by proxy at the respective meetings of your petitioners called for such purpose.

7. The meeting of petitioner AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., was duly held at Chicago, Illinois, on December 1, 1958; and the meeting of petitioner *The American Society of Refrigerating Engineers*, was duly held at New Orleans, Louisiana, on the same date. Each of said meetings was duly held upon notice, as prescribed in *Section* 43 of the Membership Corporations Law of the State of New York. The voting results at said meetings were as follows:

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.:	
In favor of the resolutions for consolidation and for the approval of the annexed Agreement for Consolidation	5,307
Against the foregoing	405
TOTAL NUMBER OF BALLOTS CAST	5,712
The American Society of Refrigerating Engineers	
In favor of the resolutions for consolidation and for the approval of the annexed Agreement for Consolidation	3,516
Against the foregoing	1,293
TOTAL NUMBER OF BALLOTS CAST	4,809

8. As shown by *Exhibit C* and the Certificate of Consolidation presented herewith, the objects and purposes of the consolidated corporation are to be as follows:

(*a*) To advance the arts and sciences of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences, for the benefit of the general public.

(*b*) To encourage and conduct scientific research and the study of principles and methods in the fields of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences, the results of which shall be made freely available to the public.

(*c*) To promote the unrestricted dissemination of knowledge and information and, for such purpose, to publish and to foster the publication of books, periodicals, papers, reports, educational programs, and scientific and educational data relating to heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences.

(*d*) To engage in educational activities (not including the conduct of any school or institution of learning), and to encourage the adoption and maintenance of high standards of instruction and educational and professional training in the fields of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences.

(*e*) To cooperate with governmental agencies and with universities, colleges, schools, and other organizations and groups having the same or similar objects and purposes, and to establish scholarships and make contributions, grants, and awards in furtherance of the foregoing purposes.

(*f*) To establish, approve, adopt, and promulgate codes, standards, and procedures in the fields of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences, subject to the proviso that all such activities shall be conducted solely— for the advancement of engineering science. -

(*g*) To organize and form local regions, chapters, sections, and student and other branches, and to regulate, operate, and control the same under the direction and at the pleasure of the consolidated corporation, but no local region, chapter, section or branch shall subject the consolidated corporation to any financial or other obligation except such as the consolidated corporation may voluntarily assume.

(*h*) To receive, acquire, hold and maintain any property, real or personal, without limitation as to amount of value, for any of the consolidated corporation's objects, by way of bequest, devise, gift, grant, purchase, or lease, to invest and reinvest the same, to control the income therefrom, and to expend or otherwise dispose of all or any portion of its funds and property, including the income, interest, or principal, subject, however, to any directions or limitations placed upon the same by donors or testators.

(*i*) To do any and all things necessary or proper in connection with or incidental to any of the foregoing.

(*j*) The consolidated corporation shall be operated exclusively for scientific and educational purposes; no substantial part of the activities of the consolidated corporation shall be the carrying on of propaganda or otherwise influencing or intending to influence legislation; in the event of the dissolution of the consolidated corporation, the board of directors shall dispose of its net assets, in trust, however, to further the purposes expressed herein, without preference in favor of any contributor or any member, officers, or director of the consolidated corporation, and subject to the order of a Justice of the Supreme Court.

The consolidated corporation is to be a non-profit corporation, and none of its activities as hereinabove set forth is to be conducted for profit; and the Certificate of Consolidation provides that no officer, director, or member of the consolidated corporation shall receive or be lawfully entitled to receive any part of the net earnings thereof or any pecuniary profit from the operations thereof, except such reasonable compensation for services in effecting one or more of its purposes as the board of directors may determine.

9. Presented herewith, incorporated herein, and separately marked *Exhibits D* and *E*, respectively, are the most recently audited financial statements of each of the petitioning corporations.

10. No previous application has been made for an order approving said Agreement for Consolidation and authorizing the filing of the Certificate of Consolidation presented herewith.

WHEREFORE, your petitioners pray for an order pursuant to *Section* 52 of the Membership Corporations Law approving the Agreement for Consolidation under the name,

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS INC., and authorizing the filing, pursuant to *Section* 50 of the Membership Corporations Law, of the Certificate of Consolidation presented herewith.

Dated, New York, January 16, 1959.

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.,
Petitioner,

By

ELMER R. QUEER (*signed*)
President

AUBRY V. HUTCHINSON (*signed*)
Secretary

The American Society of Refrigerating Engineers,
Petitioner,

By

ROBERT H. TULL (*signed*)
Vice President

ROBERT C. CROSS (*signed*)
Secretary

STATE OF NEW YORK:
SS:
COUNTY OF NEW YORK:

ELMER R. QUEER and AUBRY V. HUTCHINSON, each being duly sworn, depose and say, and each for himself deposes and says, that he, ELMER R. QUEER, is the President and he, AUBRY V. HUTCHINSON, is the Secretary of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., one of the petitioners herein; that they have read the foregoing petition and know the contents thereof; that the same is true as of their own knowledge, except as to the matter therein alleged on information and belief, and as to those matters, they believe it to be true.

ELMER R. QUEER (*signed*)

AUBRY V. HUTCHINSON (*signed*)

Sworn to before me this
16th day of January, 1959.
Mathilde B. Whitehall
Notary Public, State of New York
No. 31-4250400

STATE OF NEW YORK:
SS:
COUNTY OF NEW YORK:

ROBERT H. TULL and ROBERT C. CROSS, each being duly sworn, depose and say, and each for himself deposes and says, that he, ROBERT H. TULL, is Vice President, and he, ROBERT C. CROSS, is the Secretary of *The American Society of Refrigerating Engineers,* one of the petitioners herein; that they have read the foregoing petition and know the contents thereof; that the same is true as of their own knowledge, except as to the matters therein alleged on information and belief, and as to those matters, they believe it to be true.

ROBERT H. TULL (*signed*)

ROBERT C. CROSS (*signed*)

Sworn to before me this
16th day of January, 1959.
Mathilde B. Whitehall
Notary Public, State of New York
No. 31-4250400

EXHIBIT A

PROVISIONS OF CERTIFICATE OF INCORPORATION OF AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

1. The corporate name of the Society is to be known as AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

2. The certificate of incorporation was filed in the office of the Secretary of State on or about January 24, 1895. Amended certificates of incorporation were filed in the office of the Secretary of State on or about May 20, 1914, March 8, 1946, December 7, 1949, February 1, 1952, and December 8, 1954.

3. The number of directors shall be not less than thirteen (13) nor more than twenty-one (21). The Board of Directors shall be styled "The Council".

4. The purposes for which this corporation is organized are:

(a) To advance the arts and sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, for the benefit of the general public.

(b) To encourage and conduct scientific research and the study of principles and methods in the fields of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, the results of which shall be made freely available to the public.

(c) To promote the unrestricted dissemination of knowledge and information and, for such purpose, to publish and to foster the publication of books, periodicals, reports, educational programs and scientific and technical data relating to heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

(d) To engage in educational activities (not including the conduct of any school, schools or institutions of learning), and to encourage the adoption and maintenance of high standards of instruction and technical and professional training in the fields of heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

(e) To cooperate with governmental agencies and with universities, colleges, schools and other organizations and groups having the same or similar objects and purposes, and to establish scholarships and make contributions, grants and awards in furtherance of the foregoing purposes.

(f) To assist in the formation of local chapters and student and other branches, and to regulate, operate and control the same under the direction and at the pleasure of the corporation, but no local chapter or branch shall subject the corporation to any financial or other obligation except such as the corporation may voluntarily assume.

(g) To receive, acquire, hold and maintain any property, real or personal, without limitation as to amount or value, for any of the corporation's objects, by way of bequest, devise, gift, purchase or lease to invest and reinvest the same, to control the income therefrom, and to expand or otherwise dispose of all or any portion of its funds and property, including the income, interest or principal.

(h) To do any and all things necessary or proper in connection with or incidental to any of the foregoing.

(i) This corporation shall be operated exclusively for scientific and educational purposes; no substantial part of the activities of this corporation shall be the carrying on of propaganda, or otherwise influencing or intending to influence legislation; in the event of the dissolution of the corporation, the Council (directors) shall dispose of its net assets, in trust, however, to further the purposes expressed herein, without preference in favor of any contributor or any member, officer or director of the corporation.

5. The principal place of business or office of said Society is to be in the City and County of New York.

(a) The territory in which the corporation's operations are principally to be conducted is in all parts of the United States and its possessions and in the Dominion of Canada, but the corporation may on occasions extend its operations to other parts of the world.

(b) No officer, director or member of this corporation shall receive or be lawfully entitled to receive any part of the net earnings thereof or any pecuniary profit from the operations thereof, except such reasonable compensation for services in affecting one or more of its purposes as the Board of Directors may determine.

(c) Each director of the corporation shall be indemnified by the corporation against expenses actually and necessarily incurred by him in connection with the defense of any action, suit or proceeding in which he is made a party by reason of his being or having been a director of the corporation, except in relation to matters as to which he shall be adjudged in such action, suit or proceeding to be liable for willful negligence, misfeasance or misconduct in the performance of his duties as director; such right of indemnification shall not be deemed exclusive of any other right to which he may be entitled under any by-law, agreement, vote or otherwise.

EXHIBIT B

PROVISIONS OF CERTIFICATE OF INCORPORATION OF *The American Society of Refrigerating Engineers*

1. The name of the corporation is *The American Society of Refrigerating Engineers*, and the name has never been changed.

2. The certificate of incorporation was filed in the office of the Secretary of State of the State of New York on the 30th day of August, 1905, and in the office of the County Clerk of New York County on the 31st day of August, 1905, and a certificate of amendment for the elimination of certain provisions and purposes in the certificate of incorporation and increase in the number of trustees, directors or managers of the corporation was filed in the office of the Secretary of State of the State of New York on the 26th day of November, 1945, and in the office of the County Clerk of New York County on the 6th day of December, 1945, and a certificate of amendment for the elimination of the provision fixing the date of the annual meeting was filed in the office of the Secretary of State of the State of New York on the 23rd day of November, 1955, and in the office of the County Clerk of the County of New York on the 6th day of December, 1955, and a certificate of amendment of purposes and provisions, pursuant to *Section 30* of the Membership Corporations Law was filed in the office of the Secretary of State of the State of New York on the 28th day of December, 1956.

3. The number of directors shall be not less than fourteen (14) nor more than forty (40). The Board of Directors shall be called "The Council."

4. The purposes for which this corporation is organized are:

(*a*) To advance the arts and sciences of refrigeration, air conditioning, and the allied arts and sciences, and to elevate the standards of integrity, honor and courtesy in our engineering profession, and to foster such kindred scientific purposes as are stated herein for the benefit of the general public.

(*b*) To encourage and conduct scientific research and the study of principles and methods in the fields of refrigeration, air conditioning, and the allied arts and sciences, the results of which shall be made available to the public.

(*c*) To establish, approve, adopt and promulgate standards, codes, or recommended practice in the fields of refrigeration, air conditioning, and the allied arts and sciences.

(*d*) To promote the unrestricted dissemination of knowledge and information, and for such purpose to publish and to foster the publication of books, periodicals, reports, educational programs, and scientific and technical data relating to refrigeration, air conditioning, and the allied arts and sciences.

(*e*) To encourage educational activities (not including the conduct of any school, schools or institutions of learning), and the adoption and maintenance of high standards of instruction and technical and professional training in the fields of refrigeration, air conditioning, and the allied arts and sciences.

(*f*) To cooperate with governmental agencies and with universities, colleges, schools and other organizations and groups having the same or similar objects and purposes, and to establish scholarships and make contributions, grants and awards in the furtherance of the foregoing purposes.

(*g*) To assist in the formation of local Sections or other groups, and to regulate, operate, and control the same under the direction and at the pleasure of the corporation; but no local Section or group shall subject the corporation to any financial or other obligation except such as the corporation may voluntarily assume.

(*h*) To receive, acquire, hold and maintain any property, real or personal, without limitation as to amount or value, for any of the corporation's objects, by way of bequest, devise, gift, purchase or lease, to invest and reinvest the same, to control the income therefrom, and to expend or otherwise dispose of all or any portion of its funds and property, including the income, interest or principal.

(*i*) To do any and all things necessary or proper in connection with or incidental to any of the foregoing.

(*j*) This corporation shall be operated for scientific purposes; no substantial part of the activities of this corporation shall be the carrying on of propaganda or otherwise influencing or intending to influence legislation; in the event of the dissolution of the corporation, the Council (board of directors) shall dispose of its net assets, in trust, however, to further the purposes expressed herein, without preference in favor of any contributor or any member, officer, or director of the corporation.

5. The principal place of business or office of said Society is to be in the City, County and State of New York.

(a) The territory in which the corporation's operations are principally to be conducted is in all parts of the United States and its possessions and in Canada, and the corporation may extend its operations to other parts of the world.

(b) No officer, director, or member of this corporation shall receive or be lawfully entitled to receive any part of the net earnings thereof or any pecuniary profit from the operations thereof, except such reasonable compensation for services in effecting one or more of its purposes as The Council (board of directors) may determine.

(c) Any person made a party to any action, suit or proceeding by reason of the fact that he, his testator or intestate, is or was a director, officer or employee of this corporation shall be indemnified by the corporation against the reasonable expenses, including attorneys' fees, actually and necessarily incurred by him in connection with the defense of such action, suit or proceeding, or in connection with any appeal therein, except in relation to matters as to which it shall be adjudged in such action, suit or proceeding that such officer, director or employee is liable for negligence or misconduct in the performance of his duties. Such right of indemnification shall not be deemed exclusive of any other rights to which such director, officer or employee may be entitled by law.

Editor's Note: The *Exhibit C* referred to in Document II is the Agreement for Consolidation printed in full in this Chapter of TRANSACTIONS, beginning on p. 13. *Exhibits D* and *E* referred to in Document II are not in this volume of TRANSACTIONS.

III—Order of Court for Consolidation

At a Special Term, Part II, of the Supreme Court of the State of New York, held in and for the County of New York, at the County Court House, in the Borough of Manhattan, City of New York, on the 26th day of January, 1959.

PRESENT:

HONORABLE SAMUEL H. HOFSTADTER
JUSTICE

In the Matter of the Application of

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

and

The American Society of Refrigerating Engineers
Petitioners,

for an order pursuant to *Section* 52 of the Membership Corporations Law approving their Agreement for Consolidation under the name

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

and authorizing the filing of the Certificate of Consolidation pursuant to *Section* 50 of said Membership Corporations Law.

Upon reading and filing the joint petition of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and *The American Society of Refrigerating Engineers*, verified January 16, 1959, and the exhibits attached thereto and submitted therewith, including the provisions of the certificates of incorporation, as

amended, of each of the petitioners, the Agreement for Consolidation of said petitioners, made and entered into as of October 18, 1958, and the latest audited financial statements of each of said petitioners, and the proposed Certificate of Consolidation, made, executed, and acknowledged January 16, 1959, having been duly presented to the Court, and it appearing that said Certificate of Consolidation and Agreement for Consolidation have been duly approved by the affirmative votes cast by more than two-thirds of the members of each of the petitioners entitled to vote thereon, present in person or by proxy at respective meetings of the petitioners called for such purpose, held upon notice as prescribed in *Section* 43 of the Membership Corporations Law of the State of New York, and that 5,307 votes were cast by the members of petitioner, AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., for, and 405 votes against the consolidation and the approval of the Agreement for Consolidation, and that 3,516 votes were cast by the members of petitioner, *The American Society of Refrigerating Engineers*, for, and 1,293 votes against the consolidation and the approval of the Agreement for Consolidation, and the Attorney General having waived notice and a hearing thereon, and having certified no objection to the entry of this order, and the affidavit of Louis P. Eisner, sworn to January 23, 1959, having been filed with the petition showing that the Commissioner of Education, for and on behalf of The State Education Department, has consented to the provisions contained in the Certificate of Consolidation, and it appearing to the satisfaction of the Court that the petitioners have kindred purposes, that the provisions of *Section* 52 of the Membership Corporations Law have been duly complied with by the petitioners, and that the interests of the petitioning corporations and the public interest will not be adversely affected by the consolidation of the petitioning corporations;

NOW, on motion of Brodek & Eisner, attorneys for petitioner, AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and Stephan B. Vreeland, attorney for petitioner, *The American Society of Refrigerating Engineers*, it is hereby

ORDERED, that the agreement made and entered into as of October 18, 1958, for the consolidation of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and *The American Society of Refrigerating Engineers*, in the form annexed to the petition and marked *Exhibit C*, be and the same hereby is approved, and the filing of the Certificate of Consolidation, pursuant to said Agreement, be and the same hereby is authorized; and it is further

ORDERED, that upon the filing of said Certificate of Consolidation, together with a certified copy of this order as required, the consolidation of the petitioning corporations shall have the effect of consolidation as set forth and as provided in said Agreement for Consolidation and by *Section* 53 of the Membership Corporations Law of the State of New York.

Enter,

SAMUEL H. HOFSTADTER (*signed*)
Justice Supreme Court

The Attorney General raises no objection to the within application and waives the statutory notice thereof and has no objection to the entry of the foregoing order, and hereby waives notice of settlement thereof. 1/23/59

LOUIS J. LEFKOWITZ (*signed*)
Attorney General

KENNETH D. SHEARER (*signed*)
Assistant Attorney General

Affidavit

SUPREME COURT OF THE STATE OF NEW YORK, COUNTY OF NEW YORK

In the Matter of the Application of

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

and

The American Society of Refrigerating Engineers,
Petitioners,

for an order pursuant to *Section* 52 of the Membership Corporations Law approving their Agreement for Consolidation under the name

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

and authorizing the filing of the Certificate of Consolidation pursuant to *Section* 50 of said Membership Corporations Law.

STATE OF NEW YORK:
SS:
COUNTY OF NEW YORK:

LOUIS P. EISNER, being duly sworn, deposes and says:

I am a member of the firm of Brodek & Eisner, attorneys for petitioner, AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and am familiar with the facts and proceedings in the matter of this application for an order relating to the consolidation of said Society with *The American Society of Refrigerating Engineers.*

Attached hereto, made part hereof, and marked *Exhibit A* is a true copy of the consent of the Commissioner of Education, for and on behalf of The State Education Department, to the provisions contained in the Certificate of Consolidation.

Sworn to before me this
23rd day of January, 1959.
Mathilde B. Whitehall
Notary Public, State of New York
No. 31-4250400

LOUIS P. EISNER (*signed*)

Consent of Education Department

EXHIBIT A

THE UNIVERSITY OF THE STATE OF NEW YORK

STATE OF NEW YORK:
SS:
COUNTY OF ALBANY:

Consent is hereby given to the consolidation of "AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC." and "*The American Society of Refrigerating Engineers*" forming "AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC." as set forth in the annexed certificate of consolidation made under and pursuant to the provisions of *Section* 50 of the Membership Corporations Law.

This consent, however, shall in no way be construed as an approval by the Education Department, Board of Regents or Commissioner of Education of the purposes and objects of this corporation, nor shall it be construed as giving the officers or agents of this corporation the right to use the name of the University of the State of New York, Education Department, Board of Regents or Commissioner of Education in its publications and advertising matter, nor shall it be deemed to be a waiver of the approval of the Board of Regents for the conduct of a correspondence school by such corporation as provided in *Section* 5002 of the Education Law.

IN WITNESS WHEREOF, I, James E. Allen, Jr., Commissioner of Education of the State of New York, for and on behalf of the State Education Department, do hereunto set my hand and affix the seal of the State Education Department, at the City of Albany, this 22nd day of January, 1959.

(S E A L)

JAMES E. ALLEN, JR. (*signed*)
Commissioner of Education

IV—Certificate of Consolidation

Certificate of Consolidation of

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.
and *The American Society of Refrigerating Engineers*

into

AMERICAN SOCIETY OF HEATING, REFRIGERATING
AND AIR-CONDITIONING ENGINEERS, INC.

Pursuant to *Section* 50 of the Membership Corporations Law:

We, ELMER R. QUEER and AUBRY V. HUTCHINSON, being respectively the president and the secretary of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and ROBERT H. TULL and ROBERT C. CROSS, being respectively vice president and the secretary of *The American Society of Refrigerating Engineers*, do hereby certify:

1. (*a*) The names of the corporations to be included in the consolidation are AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and *The American Society of Refrigerating Engineers*.

(*b*) The certificate of incorporation of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., under the name THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, was filed in the office of the Secretary of State on January 24, 1895, and certificates of amendment thereof were filed on May 20, 1914, March 8, 1946, December 7, 1949, and February 1, 1952. On December 8, 1954, a certificate of change of name from THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS to AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., was filed pursuant to *Section* 40 of the General Corporation Law.

(*c*) The certificate of incorporation of *The American Society of Refrigerating Engineers*, was filed in the office of the Secretary of State on August 30, 1905, and certificates of amendment thereof were filed on November 26, 1945, November 23, 1955, and December 28, 1956.

2. The name of the consolidated corporation is AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

3. The territory in which its operations are principally to be conducted is in all parts of the United States, its territories and possessions, in Canada, and in any and all foreign countries and places.

4. The office of the consolidated corporation is to be located in the City, County, and State of New York.

5. The number of its directors shall be not less than twenty-four (24) nor more than forty-eight (48).

The directors and elective officers, and their respective terms of office, shall initially be as set forth in the Agreement for Consolidation, made and entered into as of October 18, 1958, and thereafter the directors and elective officers, and their respective terms of office, shall be as set forth in the by-laws of the consolidated corporation.

In furtherance and not in limitation of the powers conferred by statute, the board of directors, if the by-laws so provide, may designate one or more committees which, to the

extent provided in the by-laws of the consolidated corporation, or by resolution or resolutions of the board of directors, shall have and may exercise the powers of the board in the management of the business and affairs of the consolidated corporation that may be delegated to such committee or committees, such committee or committees to have such name or names as may be stated in the by-laws or as may be determined from time to time by resolution or resolutions of the board of directors.

The consolidated corporation may in its by-laws confer powers and authorities upon its board of directors in addition to the powers and authorities expressly conferred by the statutes of the State of New York or by this certificate. All powers of the consolidated corporation, insofar as the same may be lawfully vested in the board of directors, are hereby conferred upon the board of directors of the consolidated corporation.

Both members and directors of the consolidated corporation shall have the power to hold their respective meetings within or without the State of New York; the consolidated corporation shall have the power to maintain offices and keep the books of the consolidated corporation within or without the State of New York, at such place or places as may from time to time be designated by the board of directors.

No officer, director, or member of the consolidated corporation, shall receive or be lawfully entitled to receive any part of the net earnings thereof or any pecuniary profit from the operations thereof, except such reasonable compensation for services in effecting one or more of its purposes as the board of directors may determine.

Each director of the consolidated corporation shall be indemnified by said corporation against expenses actually and necessarily incurred by him in connection with the defense of any action, suit, or proceeding in which he is made a party by reason of his being or having been a director of the corporation, except in relation to matters as to which he shall be adjudged in such action, suit, or proceeding to be liable for willful negligence, misfeasance, or misconduct in the performance of his duties as director; such right of indemnification shall not be deemed exclusive of any other right to which he may be entitled under any by-law, agreement, vote, or otherwise.

6. The consolidated corporation shall be one of the constituent corporations, to-wit, AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and not a new corporation.

7. The terms and conditions of consolidation, in addition to those hereinabove set forth, and the mode of carrying the same into effect, are as follows:

(1) The purposes of the consolidated corporation are:

(*a*) To advance the arts and sciences of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences, for the benefit of the general public.

(*b*) To encourage and conduct scientific research and the study of principles and methods in the fields of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences, the results of which shall be made freely available to the public.

(*c*) To promote the unrestricted dissemination of knowledge and information and, for such purpose, to publish and to foster the publication of books, periodicals, papers, reports, educational programs, and scientific and educational data relating to heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences.

(*d*) To engage in educational activities (not including the conduct of any school or institution of learning), and to encourage the adoption and maintenance of high standards of instruction and educational and professional training in the fields of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences.

(*e*) To cooperate with governmental agencies and with universities, colleges, schools, and other organizations and groups having the same or similar objects and purposes,

and to establish scholarships and make contributions, grants, and awards in furtherance of the foregoing purposes.

(*f*) To establish, approve, adopt, and promulgate codes, standards, and procedures in the fields of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences, subject to the proviso that all such activities shall be conducted solely for the advancement of engineering science.

(*g*) To organize and form local regions, chapters, sections, and student and other branches, and to regulate, operate, and control the same under the direction and at the pleasure of the consolidated corporation, but no local region, chapter, section, or branch shall subject the consolidated corporation to any financial or other obligation except such as the consolidated corporation may voluntarily assume.

(*h*) To receive, acquire, hold, and maintain any property, real or personal, without limitation as to amount or value, for any of the consolidated corporation's objects, by way of bequest, devise, gift, grant, purchase, or lease, to invest and reinvest the same, to control the income therefrom, and to expend or otherwise dispose of all or any portion of its funds and property, including the income, interest, or principal, subject, however, to any directions or limitations placed upon the same by donors or testators.

(*i*) To do any and all things necessary or proper in connection with or incidental to any of the foregoing.

(*j*) The consolidated corporation shall be operated exclusively for scientific and educational purposes; no substantial part of the activities of the consolidated corporation shall be the carrying on of propaganda or otherwise influencing or intending to influence legislation; in the event of the dissolution of the consolidated corporation, the board of directors shall dispose of its net assets, in trust, however, to further the purposes expressed herein, without preference in favor of any contributor or any member, officer, or director of the consolidated corporation, and subject to the order of a Justice of the Supreme Court.

(2) Upon the filing of this Certificate of Consolidation in the office of the Secretary of State, each member of the constituent corporations shall have and possess the highest grade of membership in the consolidated corporation as was upon such filing held by him in either of the constituent corporations; and each member of the constituent corporations then entitled to vote therein shall possess the power to vote in the consolidated corporation for the election of directors and for all other purposes conferred upon voting members by the laws of the State of New York, this Certificate of Consolidation, and the by-laws.

(3) Except as provided in subdivision (5) of this Paragraph 7, upon the filing of this Certificate of Consolidation in the office of the Secretary of State, the separate existence of each constituent corporation, except the consolidated corporation, shall cease and the consolidation shall be effective, as provided in *Section* 53 of the Membership Corporations Law, and the consolidated corporation shall possess all of the powers of the constituent corporations, and shall have the powers and be subject to the duties and obligations of a membership corporation formed under the laws of the State of New York for like purposes.

(4) Upon the filing of this Certificate of Consolidation in the office of the Secretary of State, all the rights, privileges, and interests of each of the constituent corporations, all the property, real, personal, and mixed, and all the debts due on whatever account to either of them, and other things in action belonging to either of them, shall be deemed to be transferred to and vested in the consolidated corporation without further act or deed, and all claims, demands, property, and every other interest, whether vested or contingent, shall be as effectually the property of the consolidated corporation as they were of the constituent corporations, and the title to all real estate, taken by deed or otherwise under the laws of the State of New York vested in either of the constituent

corporation, shall not be deemed to revert or be in any way impaired by reason of the consolidation but shall be vested in the consolidated corporation.

(5) Except as provided by *Section* 12 of the Personal Property Law, or *Section* 113 of the Real Property Law, any devise, bequest, gift, or grant contained in the will of a person dying domiciled in the State of New York, or in any other instrument executed under the laws of said State, taking effect after such consolidation, to or for either of the constituent corporations, shall inure to the benefit of the consolidated corporation; and so far as is necessary for that purpose, or for the purpose of a like result with respect to a devise, bequest, gift, or grant governed by the law of any other jurisdiction, the existence of each constituent corporation shall be deemed to continue in and through the consolidated corporation.

(6) Upon the filing of this Certificate of Consolidation in the office of the Secretary of State, the consolidated corporation shall be deemed to have assumed and shall be liable for all the liabilities and obligations of the constituent corporations, in the same manner as if such consolidated corporation had itself incurred such liabilities and obligations.

IN WITNESS WHEREOF, we have made and subscribed this certificate this 16th day of January, 1959.

ELMER R. QUEER (*signed*)
President of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

ROBERT H. TULL (*signed*)
Vice President of *The American Society of Refrigerating Engineers*

AUBRY V. HUTCHINSON (*signed*)
Secretary of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

ROBERT C. CROSS (*signed*)
Secretary of *The American Society of Refrigerating Engineers*

STATE OF NEW YORK:
SS:
COUNTY OF NEW YORK:

On this 16th day of January, 1959, before me personally appeared ELMER R. QUEER, AUBRY V. HUTCHINSON, ROBERT H. TULL and ROBERT C. CROSS, to me known and known to me to be the individuals described in and who executed the foregoing Certificate of Consolidation, and they duly severally acknowledged to me that they executed the same.

Mathilde B. Whitehall
Notary Public, State of New York
No. 31-4250400

STATE OF NEW YORK:
SS:
COUNTY OF NEW YORK:

ELMER R. QUEER AND AUBRY V. HUTCHINSON, being severally duly sworn, depose and say, and each for himself deposes and says: That they are President and Secretary, respectively, of AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., a corporation duly organized and existing under and pursuant to *Section* 10 of the Membership Corporations Law of the State of New York; that as of December 1, 1958, said corporation had 11,956 members, of whom 9,607 were entitled to vote; that the membership roll of said corporation has not substantially changed since said date; that they have been authorized to execute and file the foregoing Certificate of Consolidation by the votes cast by more than two-thirds of the members of said corporation entitled to vote thereon, present in person or by proxy, at a meeting held upon notice as prescribed by *Section* 43 of Article VII of the Membership Corporations Law of the State of New York, at which a quorum of the members entitled to vote with respect to consolidation were present, in person or by proxy; and that the date of such meeting was December 1, 1958.

ELMER R QUEER (*signed*)
AUBRY V. HUTCHINSON (*signed*)

Sworn to before me this
16th day of January, 1959.
Mathilde B. Whitehall,
Notary Public, State of New York
No. 31-4250400

STATE OF NEW YORK:
SS:
COUNTY OF NEW YORK:

ROBERT H. TULL and ROBERT C. CROSS, being severally duly sworn, depose and say, and each for himself deposes and says: That they are Vice-President and Secretary, respectively, of *The American Society of Refrigerating Engineers*, a corporation duly organized and existing under and pursuant to *Section* 10 of the Membership Corporations Law of the State of New York; that as of December 1, 1958, said corporation had 7,208 members, of whom 5,594 were entitled to vote; that the membership roll of said corporation has not substantially changed since said date; that they have been authorized to execute and file the foregoing Certificate of Consolidation by the votes cast by more than two-thirds of the members of said corporation entitled to vote thereon, present in person or by proxy, at a meeting held upon notice as prescribed by *Section* 43 of Article VII of the Membership Corporations Law of the State of New York, at which a quorum of the members entitled to vote with respect to consolidation were present, in person or by proxy; and that the date of such meeting was December 1, 1958.

ROBERT H. TULL (*signed*)
ROBERT C. CROSS (*signed*)

Sworn to before me this
16th day of January, 1959.
Mathilde B. Whitehall
Notary Public, State of New York
No. 31-4250400

1652

No. 1652

ASHAE SIXTY-FIFTH ANNUAL MEETING, 1959

PHILADELPHIA, PENNSYLVANIA

THE Program (see p. 78) for the Annual Meeting consisted of 7 technical sessions, and since it was known that the merger with the *American Society of Refrigerating Engineers* had been voted by the membership, the meeting was the last held under the name of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS. During the course of the meeting, reports were presented by the Officers and Council; there was a report on the election of officers and on the adoption of the 5 proposed amendments to the By-Laws; appropriate resolutions concerning the meeting were adopted, and the newly elected Officers and Council members were duly installed in office.

The technical program featured a Topical Session on Comfort, 3 general technical sessions, and 3 sessions of the symposium type. Three papers were presented at the Topical Session on Comfort, a total of 12 papers during the 3 general sessions and a total of 14 papers were presented at the symposium sessions. In accordance with the established practice, the full text of the 15 papers presented at the Topical and 3 general sessions are printed in this volume of TRANSACTIONS. The text of the 14 papers presented at the symposium sessions are not included in this volume of TRANSACTIONS.

The meeting attracted a large attendance with more than 2600 members and guests registered, including 1516 members, 779 guests, and 329 ladies, from the States, Canada, Alaska, England, Germany, Italy, Sweden, and some South American countries.

All events on the technical program took place at the Bellevue Stratford Hotel and reflected the careful work which had been done in planning by the Committee on Arrangements of the Philadelphia Chapter.

To round out the full program, the Committee on Arrangements of the Philadelphia Chapter had included many interesting events, including a series for ladies, featured by bus tours and a luncheon and fashion show held at the Warwick Hotel on Wednesday at noon.

Held at the same time as the 65th Annual Meeting was the 14th International Heating and Air-Conditioning Exposition, under the management of the International Exposition Company, New York, N. Y., with E. K. Stevens, manager. The Exposition, held at the Convention Hall, opened on Monday at noon and remained open until 10:00 p.m. on Thursday, January 29th. It was participated in by 475 exhibitors with a total attendance reported as 20,700.

As an opening feature of the meeting, the Philadelphia Chapter had arranged for a Welcome-Pennsylvania Dutch Koffee Klatch at the Bellevue Stratford on Sunday afternoon, and in the evening a series of motion pictures was displayed in the Clover Room of the Hotel.

A Welcome Luncheon held on Monday noon was made the occasion for the presentation of the certificates to the newly elected Fellows as follows: M. F. Blankin, Philadelphia, Pa.; B. H. Jennings, Cleveland, Ohio; R. C. Jordan, Minneapolis, Minn.; B. H. Spurlock, Jr., Boulder, Colo.; and Reg. F. Taylor, Houston, Tex.

At that same luncheon the 3rd presentation of the ASHAE-Homer Addams Award was made to James Y. Payton, Los Angeles, Calif. Present to make the award was Paul K. Addams, New York, N. Y., and the presentation was made doubly interesting by the fact that Mrs. Homer Addams, widow of the donor of the award, graced the occasion with her presence.

During this luncheon E. K. Campbell, Kansas City, Mo., secretary of the Life Members Club, presented the 1959 Award of Merit, a plaque given by the Life Members Club, to Merl Baker, University of Kentucky, Lexington, Ky.

The principal speaker at the luncheon was Dr. James Creese, president, Drexel Institute of Technology, Philadelphia, who spoke on *Independence Hall and the Kremlin.*

Serving as toastmaster at the 65th annual banquet was A. J. Nesbitt, Philadelphia, Pa. The banquet was made the occasion for the introduction and installation of the newly elected Officers for the year 1959. During the installation it was pointed out that due to the merger with *ASRE* most of these Officers would not serve the full term. This was particularly true of the newly elected president, A. J. Hess, Los Angeles, Calif., who, in accordance with the plan of consolidation became First Vice President of the AMERICAN SOCIETY OF HEATING REFRIGERATING AND AIR-CONDITIONING ENGINEERS the following morning.

At the annual banquet the 17th presentation of the F. Paul Anderson Medal was made with Prof. C. P. Yaglou, Cambridge, Mass., as the recipient. Also, in accordance with the usual custom at the banquet, Past Pres. P. B. Gordon presented retiring president, E. R. Queer, with the Past President's Award.

The invocation at the banquet was by Dr. William D. Powell, general secretary, Greater Philadelphia Council of Churches. The featured speaker was Dr. Eric A.

Walker, president, The Pennsylvania State University, who spoke on *The Morning After for Education*, whose speech is printed in full in this volume of TRANSACTIONS.

Other program social features included a get-together party at the Sheraton Hotel on Monday evening and an Auf Wiedersehen Coffee Hour with the Philadelphia Chapter as hosts on Thursday morning at the hotel.

Important events of the meeting were the meetings of the Council and the Committee on Research held in the hotel on Sunday preceding the opening of the meeting. Scheduled also were various meetings of Special Council Committees and subcommittees of the Committee on Research, which were held both Saturday and Sunday preceding the meeting. Monday afternoon and Tuesday afternoon and evening were given over to meetings of many of the Technical Advisory Committees, including those on Air Cleaning, Air Distribution, Heat Transfer through Fenestration, Hot Water and Steam Heating, Odors, Heat Pump, Heating and Air-Conditioning Loads, Insulation, Physiological Research and Human Comfort, Sound and Vibration Control, Thermal Circuits, Combustion, Control, Industrial Environment, Solar Energy Utilization, Sorption, and on Weather Data.

The program was an exceptionally full one since it was an occasion not only of the usual program features of the Society, but also because numerous meetings of joint groups and committees from both the ASHAE and *ASRE* were held during the course of the meeting.

FIRST SESSION, MONDAY, JANUARY 26, 9:00 A.M.

The session was called to order by Pres. E. R. Queer, University Park, Pa. He at once recognized E. K. Wagner, Philadelphia, Pa., general chairman of the Committee on Arrangements of the Philadelphia Chapter. Mr. Wagner expressed the pleasure of the Philadelphia Chapter in having the 65th Annual Meeting held in Philadelphia and welcomed those in attendance. He also outlined the principal features of the program for the meeting which had been prepared by the committee.

President Queer then presented the Report of the President which is recorded here in full.

REPORT OF PRESIDENT

The normal functions of the Society were carried out under direction of the Council which determines the policy and the headquarters in New York City under the administrative direction of the executive secretary. The research activities under the guidance of the Committee on Research were administered by the director of research from the Research Laboratory in Cleveland, Ohio.

The major occurrence this year was the development of a plan for merging the Society with *The American Society of Refrigerating Engineers* and the presentation of the Plan to the membership of both Societies. The Plan was developed by a combined Committee on Cooperation under the guidance of the two Councils. The Society approved the Plan including a new set of By-Laws and the Consolidation Agreement on December 1, 1958, in a Special Meeting held in Chicago, Ill., by a majority of 93 percent of those submitting their proxies. In a similar action at New Orleans on the same date the Plan was approved by *ASRE* by a 73 percent majority of those submitting their proxies. These majorities were substantial considering the large response of the members expressing their desires and gratifying to those responsible for the preparation of the Plan and those who presented it to the members. The legal steps are now being processed and it

is hoped the merger will become effective at the close of this Annual Meeting or shortly thereafter. The merged Society will be known as the AMERICAN SOCIETY OF HEATING REFRIGERATING AND AIR-CONDITIONING ENGINEERS (ASHRAE). It will have its own magazine to be known as the AMERICAN SOCIETY OF HEATING REFRIGERATING AND AIR-CONDITIONING ENGINEERS JOURNAL. The other publications will be amalgamated.

Owing to the fact that the merger was of such importance, many major decisions concerning Society matters were held in abeyance, because of the effect the merger would have on the decisions concerning them. Task groups and combined committees from the two Societies are now actively engaged in resolving the details of consolidation and plans on the items held in abeyance. For example, a new research facility is now in the active planning stage and an expanded research program is being set forth.

The Society showed a net gain of 374 members and as of December 31, 1958 the total membership stood at 12,087. Attention is directed to the fact that 1312 applications were processed, whereas the net gain was only 374 members. This indicates a casualty of 938 members in spite of the special efforts put forth by the regional directors and chapters to reduce this number. Special attention must be given this matter and means found to hold the casualty list to a minimum.

The Net Worth of the Society was $524,448 and this represents slightly over $43 per member. As usually occurs in a year without an Exposition there is a decrease in the Net Worth. The budgeted expenditures for services to the membership were slightly over $58 per member. The financial condition is strong.

An International Affairs Committee was activated this year and it arranged for a visitation of a group of West German engineers to come to this country under the sponsorship of the Society. There are at this Annual Meeting similar groups from Sweden and France.

Three new chapters were organized and installed. They are located in Calgary, Alberta, Canada; San Diego, Calif.; and Providence, R. I.

President Queer also announced that the following were appointed as members of the Committee on Resolutions and that a report from the committee should be available for presentation at the closing session of the meeting: S. F. Gilman, Syracuse, N. Y., chairman; C. B. Gamble, New Orleans, La., and W. J. Olvany, New York, N. Y.

The next order of business was the Treasurer's Report by J. H. Fox, Toronto, Ont., Canada, who, upon being recognized by President Queer, presented the following report, and since the Accountant's Report is a part of the Treasurer's Report, excerpts from the Accountant's Report are presented here:

TREASURER'S REPORT

During the fiscal year, ending October 31, 1958, the Society had a gross income of $578,956.77 and total expenditures of $656,468.94—an excess of expenditures over income resulting in a deficit of $77,512.17.

$58,092.15 of the total deficit was incurred by excess Research Expense over Research Income and charged to the Research General Fund; $19,420.02 of the total deficit was incurred by excess of Society expense over General Society Income and was charged to the Society General Fund.

Factors contributing to this heavy deficit are:

1. Decline in research contributions	$ 56,130.86
Increase in dues and other income	21,390.25
Resulting in total net decline in income	$ 34,740.61

2. Increase in expenses were incurred in accounts covering:
   (a) Committee and Chapters
   (b) GUIDE Publication
   (c) Headquarters Operations—general increase in staff salaries alone was $33,378.55.
   (d) Building Reserve
   (e) Contributions

For total net increase	$ 29,633.57
Total effect of Items 1 & 2	$ 64,374.18

It should be noted that the 1958 Budget forecast a deficit as follows:

(a) Society General	$ 18,465.00	
(b) Research	48,190.00	
Total forecast deficit		$ 66,655.00
Compared against actual deficit:		
(a) Society General	$ 19,420.02 for variance of	$ 955.02
(b) Research	58,092.15 for variance of	9,902.15
Total variance from budget		$ 10,857.17
		1.54%

INCOME

Dues	$170,051.58	29.3 %
Publications	226,910.57	39.2
Research contributions (dues portion included)	169,413.38	29.3
Investments	493.02	0.1
Admission fees, certificates	11,830.25	2.05
Miscellaneous	257.97	0.05
	$578,956.77	100.0 %
Transferred from reserves	73,000.00	
	$651,956.77	

Of the above total, it will be noted that there was allocated to research:

From dues (40 percent of Members, Associates over 30, and Affiliates)	$108,237.39
Contributions—general	20,749.09
—earmarked (less deferred)	24,431.19
—contracts (U. S. Navy)	15,403.35
Interest on investments	592.36
	$169,413.38
Transferred from reserves	48,000.00
Total research allocation	$217,413.38

Funds available for research were 29.3 percent of current Society receipts. Funds available from total receipts and reserves—33.3 percent.

EXPENDITURES

The total expenditures of $656,468.94 were made up as follows:

Committees and Chapters	$ 39,336.81	6.0%
National Meetings	8,491.32	1.3
Publications—JOURNAL, TRANSACTIONS, GUIDE	140,912.60	21.5
Headquarters	240,222.68	36.6
Research	227,505.53	34.6
	$656,468.94	100.0%

BALANCE SHEET

At the close of the fiscal year, the Society had assets in the form of cash, securities, certain real estate and properties in Cleveland, plus capital equipment at New York headquarters and the Research Laboratory. The book values of these assets were as follows:

Property Assets:		
1. Buildings depreciated from cost	$ 90,207.75	
2. Laboratory equipment depreciated at 10 percent per year	21,771.82	
3. Furniture & Fixtures (New York) depreciated at 10 percent per year	12,400.79	
	$124,380.36	
Building Reserve Fund and Maintenance Fund	42,506.32	$166,886.68
General Funds		193,403.84
Reserve Funds and Investments		155,748.59
Research Funds		168,166.41
Pension Reserve (set up 1958)		1,395.38
Miscellaneous Trust Funds		18,672.84
		$704,474.74
Miscellaneous Liabilities		180,026.38
NET WORTH OR BALANCE (In allocated trusts, building funds, etc.)		$524,448.36

Within the liabilities of the Society is an item of $50,701.19 due to the Research General and Research Reserve Funds.

Within the present method of accounting, the net worth of the Society has been reduced from $582,963.50 at the end of fiscal 1957 to $524,448.36 at the end of fiscal 1958—a net reduction of $58,515.14. It is noted that the general funds available for general operation of the Society and Research Laboratory were at an impractical low level of $13,377.46 while both the General and Research Reserve Accounts have increased materially. In the general accounts there has been a reduction of $86,270.86 while at the same time other general reserves have increased $20,814.07 or a net cash operating deduction of $65,456.79.

This condition has necessitated considerable borrowing from within the various reserve accounts for operating purposes, all in anticipation of future income. The situation is reflected within the Liabilities Statement to the amount of $98,701.19—so that the liabilities to various trade accounts, for salaries, and employees deferred income taxes and similar items is $91,325.19.

CONCLUSIONS

The statement under review reflects the results and effects of:

1. Increases in staff additions, salaries, and general increases in costs of doing business or operating a Society.

2. The first funding and establishment of an employees pension fund.

3. The very favorable results obtained from the assistance rendered by investment counsel (Bankers Trust Company), to obtain maximum investment return—plus the value of obtaining the maximum capital growth within the fullest terms of investment security.

4. The severe reduction in the amount of research contributions as against the anticipated amount has caused a heavier than normal demand on reserves. The Committee on Research is alert to this situation and is taking steps to alleviate the problem and at the same time not prejudice the essential contribution made by that phase of the Society activities.

It may be stated that the main purposes of reserves have been justified in this past year in that they have served a real purpose to tide the operation through a period of adjustment—during the time of national uncertain economic conditions.

There is some question as to the continuance of the fixed amount of 40 percent of dues to research. Full consideration should be made as to the possibility of making the percentage of this allotment more flexible.

5. Certain extraordinary non-recurring expenses were incurred due to extra work load in regard to the anticipated merger with *ASRE*.

6. It is obvious that 1958 being a year in which no income was derived from Exposition contributions, current financing difficulties were increased.

7. In view of the fact that merger with *ASRE* is now approved and is practically certain of conclusion early in 1959—no recommendations are made for future financial operations. A completely new set of conditions can be anticipated. In general, the finances of the Society are in good health and reflect a long period of conservative management.

Any recommendations at this time would be in regard only to details of operation and do not concern any basic policies.

Accountant's Report

FRANK G. TUSA & CO.

Certified Public Accountants

50 Broad St., New York 4, N. Y.

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.
62 WORTH STREET, NEW YORK, N. Y.

Gentlemen:

In accordance with the authority contained in the minutes of the Council meeting on January 30, 1958, we have examined the books of account of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.—New York and Cleveland for the fiscal year ended October 31, 1958.

Our examination was conducted in accordance with generally accepted auditing standards, and accordingly included such tests of the accounting records and such other auditing procedures as we considered necessary in the circumstances.

We are pleased to present our report of this examination which includes the various financial schedules and statements listed in the table of contents. Your attention is directed to the following comments which further define the scope of our examination, and which we consider to be of general interest and assistance in the interpretation of the results of the Society's activities for the year ended October 31, 1958.

COMMENTS

The Society's activities for the current fiscal year resulted in a funded deficit of $77,512.17 as compared to a budgeted deficit of $66,655.00 revealing an over all budgetary deviation of 1.54 percent.

A comparison of the operations for the fiscal years 1958 and 1957 shows a deficit of $77,512.17 in 1958 against a deficit (after excluding Exposition Contributions) of $13,137.99 in 1957. A summary of the factors contributing to this change is as follows:

Decline in contributions received	$56,130.86	
Less: Net increase in Dues and Other Income of	21,390.25	
Decline in Gross Revenues		$34,740.61
Increase in expenses (of which salaries amounted to $33,378.55) of		29,633.57
Resulting in a Comparative Unfavorable Change of		$64,374.18

Included within the statement of changes in Fund Balances is an appropriation of $1,595.38 representing a funded reserve of one year's contribution for the Special Pension Agreement for employees not currently participating in the Society's standard retirement plan. This funded contribution is in accordance with the directions contained in the minutes of the meeting of the Council on October 18, 1958.

The following investment ratios signify the long-term financial condition of the Society.

OF THE NET WORTH OF THE SOCIETY $524,448.36, there is:

a) Invested in Government securities	18.59%
b) Invested in Non-Governmental Securities	18.00%
c) Total investment in all securities	36.59%

AS TO TOTAL INVESTMENTS IN ALL SECURITIES

a) Invested in Government securities	50.81%
b) Invested in Non-Governmental Securities	49.19%
	100.00%

CASH—PAGE 12*	October 31, 1958	1957
On Hand	$ 150.00	$ 150.00
Investment Advisory Account	2,542.89	284.26
Special Pension Account	1,595.38	–0–
Checking Accounts	46,568.13	76,992.24
Savings Bank	156,630.67	214,245.31
	$207,487.07	$291,671.81

All cash was verified either by count or by direct communication with the depositories. All balances reported to us were reconciled to those reflected on the books of account and were correct. Every depository responded. A detailed listing of cash balances is to be found on page 12 and a summary schedule accounting for the decrease in the cash position is to be found on page 25.

DUES AND ACCOUNTS RECEIVABLE

The following classification of Dues and Accounts Receivable were submitted to us by the internal accounting staff:

MEMBERSHIP

Members	$4,293.85	
Associate Members (over 30)	6,720.62	
Affiliates	4,401.25	
Associate Members (under 30)	437.75	
Students	12.00	$15,865.47

* Page numbers refer to Table of Contents in Accountant's Report submitted.

SUNDRY DEBTORS	9,075.47
	$24,940.94

Our examination included a review of the internal control utilized in the accounting for the above receivables, and in our opinion found it to be adequate. We requested outside confirmation of sundry Debtors but not of Dues Receivable. We were, however, through other auditing procedures able to confirm substantively the Dues Receivable balances, which we found to be correct. In determining the adequacy of the allowances for dues and accounts doubtful of collection, we reviewed the budgetary provisions for the fiscal year ended October 31, 1959, and such other pertinent data as we considered necessary. In our opinion, the allowances for these doubtful accounts will be ample in absorbing possible losses due to non-collection for the budgeted fiscal year 1959.

INVESTMENTS

Confirmation was received from the Bankers Trust Company of the investment portfolio held for the credit of the Society at October 31, 1958. A statement of this account's financial condition and changes therein is to be found on pages 13, 14, and 15 of this report.

INVENTORIES

The following inventories were verified either by physical count or by direct communication with the printers.

TRANSACTIONS	$ 5,051.32
Emblems	440.36
GUIDE Paper	12,800.69
	$18,292.37

A schedule of TRANSACTIONS inventories follows:

VOLUME	YEAR	QUANTITY	COST OF EACH	AMOUNT
1–57	1895–1951	5,671	0.4000	$2,268.40
58	1952	122	1.7960	219.11
59	1953	394	2.0000	788.00
60	1954	155	2.0665	320.31
61	1955	117	2.0742	242.68
62	1956	100	2.1273	212.73
63	1957	438	2.2833	1,000.09
				$5,051.32

ADVANCE DEPOSIT—AIR TRAVEL	$ 425.00

This deposit placed with the United Airlines was confirmed by direct communication.

PROPERTY FUND ASSETS (PAGE 11)	$124,380.36

All assets reflected within this fund are at cost of acquisition. With the exception of Buildings, all assets have been depreciated at the rate of 10 percent per year. Depreciation in the amount of $11,560.51 has been funded in the current fiscal year while expenditures for new assets and capital improvements amounted to $17,258.68.

DEFERRED INCOME

The membership classification of the dues prepaid by elected members follows:

MEMBERSHIP	
Members	$ 15,443.59
Associate Members (over 30)	8,006.83
Affiliates	9,781.25
Associate Members (under 30)	1,933.25
Students	75.00
	$ 35,239.92

FUNDS

There is included on Page 16 a statement of the changes in the fund's balances which occurred during the fiscal year ended October 31, 1958.

INSURANCE

The insurance coverage of the Society follows:

FIRE

Building (Cleveland)		$201,000.00
Personal Property:		
Cleveland	$104,000.00	
Printers	17,000.00	
New York	26,000.00	147,000.00

SPRINKLER LEAKAGE

Waverly Press	10,000.00	
Horn-Shafer Co.	5,000.00	15,000.00

NON-OWNERSHIP—AUTOMOBILE—COMPREHENSIVE

Public Liability	100/300M
Property Damage	25M
Medical	1M

GENERAL PUBLIC LIABILITY—PREMISES

New York and Cleveland	
Liability	100/300/300M
Bodily Injury	25/100/100/0/100M

COMMERCIAL BLANKET BOND

President, Treasurer, Chairman of the Finance Committee, and First and Second Vice-President	20,000.00

CONCLUSION

In our opinion, the accompanying statements of financial condition and of income and expenses and fund transactions of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., present fairly its financial condition at October 31, 1958 and the results of its operations for the year then ended, in conformity with generally accepted accounting principles applied on a basis consistent with that of the preceding year.

FRANK G. TUSA & CO.,
CERTIFIED PUBLIC ACCOUNTANTS

November 24, 1958

SPECIAL COMMENT

In reading and examining that part of this report which compares the budget as adopted by the Council, and the actual cash transactions of the Society as a whole, we should like to bring to the attention of the reader that those schedules reflect only the flow of cash in relation to the Society's fiscal operations.

The figures representing budgetary comparisons are not to be interpreted as being synonomous with any statement or schedule preceding them within this report.

The statements of Financial Condition, of Income and Expense and of Funds presented herein reflect the financial condition and operations inclusive of non-cash auditing adjustments and cash items not yet received but accrued by us as receivables or payables, with their counterpart measured in income and/or expense.

STATEMENT OF FINANCIAL CONDITION—October 31, 1958

ASSETS

GENERAL FUNDS				
Society—New York			$110,865.48	
Research—Cleveland			82,538.36	$193,403.84
RESERVE FUNDS				
Society—New York				
Cash on Deposit		$ 32,141.49		
Investments				
Undivided interest in Investment Advisory Account	$129,667.64			
Less: Allowance for Decline in Market Value thereon	6,060.54	123,607.10	155,748.59	
Research-Cleveland				
Cash on Deposit		95,166.41		
Due from Society General Fund	25,000.00			
Due from Research General Fund	48,000.00	73,000.00	168,166.41	323,915.00
CAPITAL ASSET FUNDS				
Property Fund			124,380.36	
Building Fund				
Cash on Deposit		464.06		
Undivided interest in Investment Advisory Account	42,238.00			
Less: Allowance	1,974.15	40,263.85	40,727.91	
Building Maintenance Reserve Fund				
Cash on Deposit			1,779.41	166,887.68
SPECIAL FUND				
Special Pensions Fund				
Cash on Deposit				1,595.38
ENDOWMENT FUNDS				
F. Paul Anderson Award				
Cash on Deposit		378.47		
Undivided interest in Investment Advisory Account	1,038.48			
Less: Allowance	48.54	989.94	1,368.41	
Educational Fund-Life Members				
Cash on Deposit			78.97	
Research Endowment Fund				
Cash on Deposit			833.79	
Gustus L. Larson				
Cash on Deposit		113.77		
Undivided interest in Investment Advisory Account		6,074.68	6,188.45	
Homer Addams				
Cash on Deposit		303.79		
Undivided interest in Investment Advisory Account	10,384.81			
Less: Allowance	485.38	9,899.43	10,203.22	18,672.84
				$704,474.74

LIABILITIES AND FUNDS

LIABILITIES				
GENERAL FUNDS				
Society—New York				
Employees' Income Taxes Withheld				
Federal	$ 1,958.62			
N. Y. State	1,119.05	$ 3,077.67		
Due to Research General Fund	25,701.19			
Due to Research Reserve Fund	25,000.00	50,701.19		
Accrued Accounts Payable				
Trade Creditors	12,800.69			
Salaries	15,202.50			
Professional Fees	1,750.00			
Junior Student Member Award	100.00			
Social Security Taxes	114.46	29,967.65		
Prepaid Dues and Fees				
Candidates Dues	597.75			
Admissions Fees	2,639.28			
Cash on Deposit-Per Contra		3,237.03		
Deferred Income				
Elected Members' Dues		21,947.25		
Reserve for Fluctuation-Canadian Exchange		61.86		
TOTAL SOCIETY GENERAL FUND LIABILITIES			$108,992.65	
Research—Cleveland				
Accounts Payable				
Hospitalization	111.00			
Fire Insurance—Long Terms	6,497.24	6,608.24		
Due to Research Reserve Fund		48,000.00		
Deferred Income				
Elected Members' Dues	13,292.67			
Unexpended Earmarked Contributions	3,132.82	16,425.49		
TOTAL RESEARCH GENERAL FUND LIABILITIES			71,033.73	
TOTAL LIABILITIES				$180,026.38
FUNDS				
Society General—New York		1,872.83		
Research General—Cleveland		11,504.63	13,377.46	
Society Reserve—New York		155,748.59		
Research Reserve—Cleveland		168,166.41	323,915.00	
Property		124,380.36		
Building		40,727.91		
Building Maintenance Reserve		1,779.41	166,887.68	
Special Pensions Fund			1,595.38	
F. Paul Anderson Award		1,368.41		
Educational Fund—Life Members		78.97		
Research Endowment		833.79		
Gustus L. Larson		6,188.45		
Homer Addams		10,203.22	18,672.84	524,448.36
				$704,474.74

CONDENSED COMPARATIVE STATEMENT OF INCOME AND EXPENSES

	OCTOBER 31, 1958	1957	COMPARATIVE CHANGES
INCOME			
Society			
Dues	$170,051.58	$161,353.67	$ 8,697.91
Admission Fees	10,660.00	11,555.00	(895.00)
Emblems, Pins, Certificates	1,170.25	1,174.25	(4.00)
Publications			
JOURNAL Contract	30,000.00	30,000.00	-0-
THE GUIDE	178,974.63	186,072.14	(7,097.51)
TRANSACTIONS	1,889.69	2,291.65	(401.96)
Books, Reprints, Codes, etc.	12,388.85	7,437.98	4,950.87
Membership Roll	3,657.40	6,010.05	(2,352.65)
Exposition Contribution (*Note A*)*	-0-	-0-	-0-
Investments	493.02	621.55	(128.53)
Foreign Exchange	257.97	488.34	(230.37)
Miscellaneous	-0-	627.33	(627.33)
Research			
Dues	108,237.39	99,967.11	8,270.28
Contributions—General	20,749.09	21,913.60	(1,164.51)
Contributions—Earmarked	24,431.19	80,562.05	(56,130.86)
Interest	592.36	900.94	(308.58)
U. S. Navy—Bureau of Ships	15,403.35	2,721.72	12,681.63
TOTAL INCOME	578,956.77	613,697.38	(34,740.61)
EXPENSES			
Committees and Chapters	39,336.81	32,120.74	7,216.07
Meetings	8,491.32	11,429.22	(2,937.90)
Publications			
Members' Subscriptions to *HPAC*	563.95	526.36	37.59
TRANSACTIONS	26,231.16	26,957.80	(726.64)
Membership Roll, Books, Reprints, Codes, etc.	7,917.76	13,635.60	(5,717.84)
THE GUIDE	106,199.73	100,356.12	5,843.61
Headquarters	240,222.68	233,192.68	7,030.00
Research			
Promotion—Fund Raising	9,802.28	16,693.67	(6,891.39)
Committee Expenses	1,877.84	3,739.79	(1,861.95)
Staff Salaries	141,145.19	107,766.64	33,378.55
Building Operation and Maintenance	11,686.13	11,620.72	65.41
Provision for Building Maintenance Reserve	2,500.00	2,000.00	500.00
Cooperative Research	23,274.02	38,661.25	(15,387.23)
Laboratory	35,920.07	27,464.49	8,455.58
Contributions	1,300.00	-0-	1,300.00
	656,468.94	626,165.08	30,303.86
NET OPERATING REVENUES	(77,512.17)	(12,467.70)	(65,044.47)
OTHER DEDUCTIONS			
Provision for Decline in Market Value of Investments	-0-	670.29	(670.29)
(DEFICIENCY) OF INCOME OVER EXPENSES	($ 77,512.17)	($ 13,137.99)	($64,374.18)

* *Note A:* Exposition Contributions in the amount of $132,002.13 for the fiscal year ending October 31, 1957, has been excluded from here in order to reflect, more properly, the comparative changes for the fiscal years presented.

The next order of business was the presentation of the Report of the Council and President Queer called on the Executive Secretary, A. V. Hutchinson, New York, N. Y., who read the Report of the Council which is printed here in full.

REPORT OF THE COUNCIL

The 1958 Council of the Society held 6 meetings during its term, organizing for business January 30, 1958, at Pittsburgh, Pa. and concluding its work January 25, 1959, in Philadelphia. Interim meetings were held April 20, in Milwaukee, Wis., June 22, in Minneapolis, Minn., October 18, at Las Vegas, Nev., and December 1, at Chicago, Ill. The Officers were present at all meetings and Council members' attendance was 100 percent at the January and June 1958 and the January 1959 meetings.

The major matter of discussion was the proposed Merger Plan with *ASRE* which culminated in a favorable vote by the membership of both Societies on December 1, 1958. The merger agreement approved by the Councils and members of both organizations and the petition for merger were signed by the Officers of both Societies and filed with the proper New York State authorities as required by the Membership Corporations Law to form the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

The status of ASHAE membership on January 1, 1959 was as follows:

	January 1, 1958	January 1, 1959
Honorary Members	5	4
Presidential Members	20	19
Life Members	240	255
Fellows	12	9
Members	4,886	4,989
Associate Members (over 30)	3,646	3,712
Associate Members (under 30)	594	699
Affiliates	2,130	2,226
Students	180	174
	11,713	12,087
NET GAIN		374

January 1958–January 1959	
Elections & Reinstatements	1,287
Cancellations, Resignations	913
NET GAIN	374

As required by *Section* 46 of the Membership Corporations Law of the State of New York, the following financial report is presented by the Council and filed herewith:

Assets—October 31, 1958

The Society has the following Assets:

Laboratory, Property and Equipment—net	$124,380.36
Cash—New York and Cleveland	207,487.07
Investments at adjusted cost with Bankers Trust Company (Investment Account) and Accrued Interest	193,216.31
Dues and Accounts Receivable—net	15,887.04
Inventories	18,292.37
Prepaid Expenses	47,003.78

Total Assets—1958 (*Note A*)	606,266.93
Total Assets—1957	633,983.72
Decrease for the year	$ 27,716.79

Liabilities—October 31, 1958

The Society has the following liabilities:

Accounts Payable (*Note B*)	$ 20,116.77
Federal Income Taxes	3,077.67
Accrued Accounts	16,952.50
Deferred Income	
Research Projects	3,132.82
Prepaid Admission Fees, Dues	38,476.95
Reserve for Fluctuation in Canadian Exchange	61.86
TOTAL LIABILITIES	81,818.57
NET WORTH (Funds)	524,448.36
TOTAL LIABILITIES AND NET WORTH	$606,266.93

Note A—After elimination of following inter-funds Receivable and Payable

Due to Society General Fund (Investment Advisory)	$ 739.82
Due to Research General Fund	25,701.19
Due to Research Reserve Fund	73,000.00
	$99,461.01

Note B—Includes an amount of $493.38 due to Bankers Trust Company for Investment Advisory Fees reflected as an account liability of the Investment Advisory Account

Two Regional Directors were appointed for one year, Messrs. Fred Janssen and P. N. Vinther. During the year 3 new chapters were organized: Southern Alberta Chapter, Calgary, Canada; San Diego Chapter, San Diego, Calif.; and Rhode Island Chapter, Providence, R. I. Chapters Regional Committees met in their respective regions:

Region 1	New York, N. Y.	March 17
Region 2	Milwaukee, Wis.	April 21
Region 3	Moline, Ill.	May 6
Region 4	Spokane, Wash.	May 2–3
Region 5	Washington, D. C.	April 9
Region 6	Albuquerque, N. M.	May 20
Region 7	Winnipeg, Man., Canada	April 28

The Council, General, and Special Committees were appointed at the organization meeting; the Certified Public Accountant, Frank G. Tusa & Co., and Legal Counsel, L. P. Eisner, were reappointed and the official depositories for funds in New York, Cleveland and Toronto were designated; the Bankers Trust Company, New York, N. Y., was retained as Investment Advisor.

Cooperative Research Agreements were signed for special projects upon the recommendation of the Committee on Research.

A retirement policy for employees was established and special contracts for 5 staff members were authorized.

A new lease for 18 months from July 1, 1958 was required to maintain Society Headquarters at 62 Worth St., New York, N. Y.

The JOURNAL publication contract was terminated December 1958 by mutual agreement with the publishers. An interim JOURNAL for the months of January and February 1959 was authorized pending the merger.

A new award to be known as the ASHAE-Willis H. Carrier Award was announced. Awards made for 1958 were as follows:

F. Paul Anderson Medal—Prof. C. P. Yaglou, Boston, Mass.,
ASHAE-Homer Addams Award—John Y. Payton, University of California at Los Angeles (for 1959),
Life Members Award—Prof. Meri Baker, University of Kentucky.

The usual routine actions were taken by the Council as required by the By-Laws with respect to admission of members, acceptance of resignations, and cancellation of membership.

Respectfully submitted,
E. R. QUEER, *President*
JOHN H. FOX, *Treasurer*

President Queer then announced that there were certain proposed changes in the By-Laws to be considered as the next item of business. On his request, Mr. Hutchinson read the 5 proposed changes to the By-Laws. These amendments are stated here showing the revised wording proposed in each case.

ASHAE PROPOSED BY-LAW AMENDMENTS

1. **Article IV, Funds:** *Section 6.* **Investment of Funds.** (*a*) The Society Reserve Fund and such other funds as may be allocated by the Council for investment may be invested and reinvested, without restriction, in any kind of property, real, personal or mixed.

2. **Article VII, Committees:** *Section 3.* **General Committees.** (*e*) Honors and Awards Committee, consisting of three (3) MEMBERS who have been MEMBERS in good standing for not less than ten (10) years, have served on the Council, and who are well acquainted with the ideals of the Society and the activities, prominence, and prestige of its Members. Subject to the direction of the Council, the said committee shall be responsible for the administration of all Honors and Awards granted by the Society; the review, interpretation, and revision of policy with respect thereto; the acceptance of grants for the establishment of awards; and the creation of new awards which will assist in attaining the purposes and objectives of the Society.

3. **Article VII, Committees:** *Section 3.* **General Committees.** (*f*) *Chapters Regional Committees*, each serving one Regional Area, and each consisting of the Regional Director for the Area and one (1) member and one (1) alternate member selected by each Chapter therein, to serve for a term of one (1) year. The said committees shall solicit from the Chapters and Branches within their respective Regional Areas recommendations concerning the policies, procedures, and operation of the Society, its Chapters and Branches, review the same, and make recommendations thereon to the Regions Central Committee. Said committees shall select the MEMBERS and alternates to serve on the Nominating Committee, and duly notify the Executive Secretary of such selections. The alternate members of Chapters Regional Committees may be present at committee meetings and participate in the deliberations thereof, but shall not vote therein except in the absence of the committee members for whom they respectively are alternates. The said committees shall hold committee meetings prior to December 1 of each year. Each Regional Director shall be the chairman of the committee serving his Regional Area.

4. **Article VIII, Meetings, Nominations and Elections:** *Section 3.* **Nominating Committee.** . . . Alternate members may be present only at the organization meeting of the committee at the Annual Meeting but shall not participate in the deliberations thereof or vote therein. No alternate member may be present at subsequent committee

meetings except in the absence or disability or resignation of the member or the first alternate of the one of the eight (8) groups with which he was selected. The absence, disability or resignation of a member or an alternate member eligible to participate shall serve automatically to terminate his membership on said committee. . . .

5. **Article IX, Amendments:** *Section* 1. **Prerequisites.** These By-Laws may be amended by a two-thirds vote of the Society at an Annual Meeting or Semi-Annual Meeting thereof, provided that written notice of the proposed amendment. . . .

Since the next item on the program for the session was a Topical Session on Comfort with 3 technical papers to be presented, President Queer read the rules governing a session of this kind, and introduced C. S. Leopold, Philadelphia, Pa., as moderator during the presentation of the technical papers. Three papers were presented and discussed, following which Mr. Leopold returned the session to Treasurer Fox, who adjourned the session at 12:20 p.m.

SECOND SESSION, TUESDAY, JANUARY 27, 9:30 A.M.

The presiding officer at the second session was First Vice Pres. A. J. Hess, Los Angeles, Calif., who, after opening the meeting, called on E. F. Snyder, Jr., Minneapolis, Minn., chairman, Committee on Research, who presented the 1958 Annual Report of that committee. This report is here printed in full. This same report appears in the January 1959 ASHAE JOURNAL, p. 11.

ANNUAL REPORT OF COMMITTEE ON RESEARCH—1958

This brief report covers the fiscal year of the Society for the period of November 1, 1957, to October 31, 1958 During this period, research was carried out both at the Laboratory and at 10 cooperating institutions. At the start of the year, the Technical Advisory Committees were realigned to total 18. However, during the year it appeared desirable to form an additional Technical Advisory Committee on Control and this was created. The total current membership of Technical Advisory Committees is 276 and 33 meetings of these committees were held during the year. The Committee on Research and its Executive Committee held 6 meetings.

FINANCES AND EXPENDITURES

The research program of the Society obtains its support from 3 sources; the *first* of which is that fraction of the membership dues, amounting to 40 percent, which is allocated to research; the *second* is the significant contribution for research which in recent years has been made by the International Heating and Air-Conditioning Exposition; and the *third* consists of the generous contributions given to the research program by industrial firms, associations and individuals. Funds from the latter category are allocated either for general use or for the support of specific research projects or fields. Additional funds are also received for those research projects carried out for the United States Government and for foundations interested in projects of wide general interest. Even though the year 1957–58 witnessed a recession in the national economy and there were some reductions in the contributions received from industry, the Laboratory operation increased its expenditures for research work by almost $34,000, with the total amounting to $235,562. The graphs shown in Fig. 1 indicate the division of expenditures and the pattern they have followed over the last 13 years.

FACILITIES AND STAFF

The Research Laboratory, located in Cleveland on Euclid Avenue, consists of two buildings. The front building, which is used for office space, has an area of 5300 sq ft and the rear building, largely housing the research activities, has 13,000 sq ft of useable space. An improvement program on the front building, started in 1957, was completed early in 1958. This refurbishing was so vitally necessary that it was carried out even though planning is now under way which should lead to relocating the Laboratory facilities and constructing a building or buildings to provide facilities for a greatly expanded

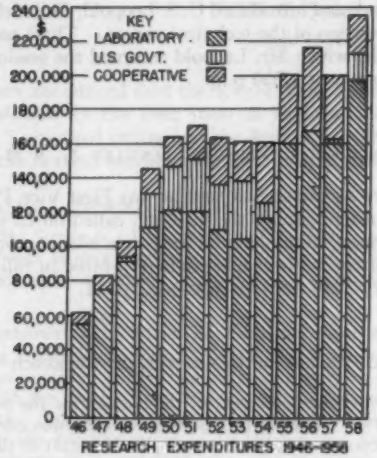

FIG. 1...CHART SHOWING THE NATURE OF SOCIETY RESEARCH EXPENDITURES FROM 1946–1958

research program. At the site in Cleveland, a third building also exists which houses the solar calorimeter. It is in this building that the solar intensity studies have been carried out with the most recent program being a study of the effectiveness of shading devices both outside and inside of building fenestration, with some of the studies being devoted to tests of between-glass shading.

Facilities at the Laboratory are very complete and adequate for the research programs which are now under way. Particular mention should be made at this time of the newly constructed Environment Laboratory. This facility is a completely-instrumented structure built inside the main Laboratory and is a flexible test center for carrying out environmental studies over a wide range of controlled conditions. The main test floor of the Laboratory has also been used during the last year for the elaborate duct lay-outs required for studies on noise related to elbows and to duct runs in general.

The Laboratory receives extensive current reference material which is carefully examined and catalogued for further reference. A library in the Laboratory is available for use by the research staff and is also available to members of the Society and to others in the profession and industry who request its services. The conference room of the library is available for meetings of Technical Advisory Committees, the Committee on

Research, Society Committees and even for outside groups having interests closely related to those of the Society.

The research work of the Laboratory is supplemented by an extensive program of cooperative research which is carried out at university and research laboratories throughout the country. Such cooperative research projects are encouraged whenever it is felt that the facilities and available staff at an outside institution have such special interest that a research project can more expeditiously be carried to completion than might be possible at the Laboratory. By using cooperative facilities, the research base of the Society is broadened as additional researchers are made available for the profession and the needs and research problems of the industry are spread over a more extended area. The programs of the cooperating institutions are closely followed by the Laboratory research staff with help and suggestions given by members of the directly interested Technical Advisory Committees. Reports of cooperative research work are all presented at Society meetings and reproduced in Society publications.

Results of research carried out at the Laboratory and at its cooperating institutions have contributed significantly to the literature in the field of heating, ventilating, and air conditioning. Research work, first presented in the JOURNAL and later in the TRANSACTIONS of the Society, ultimately find its way, in condensed and useable form, to the HEATING VENTILATING AIR CONDITIONING GUIDE published by the Society. This volume is internationally accepted as the authoritative reference book in the many fields of interest of the Society.

The research staff at the Laboratory consists of administrative staff, project engineers, assistants, a librarian, secretarial and maintenance staff. Of this number, 11 members of the staff are of professional level and the total number as of October 1958, including environmental subjects, total 27. The Laboratory was fortunate during the year to obtain the services of Walter Koch, M.D., PhD. Dr. Koch's functions with the Laboratory are to advise and guide both on the subjects and on the studies being carried out in the Environment Laboratory.

The remainder of this report summarizes, in detail, activities carried out at the Laboratory and in cooperating institutions. Because this work is largely associated with Technical-Advisory-Committee activities, the individual topics are listed in relation to each Technical Advisory Committee.

TAC ACTIVITIES

During the year, a conference was held at the Laboratory on May 14, 1958, to consider broad aspects of the whole field of Control. This conference brought out a number of interesting points and indicated definitive needs for further studies in this field. One recommendation which came from the conference was that a Technical Advisory Committee on Control be formed and in accordance with the recommendation, the chairman of the Committee on Research appointed the committee. Its report is included with those of the other Technical Advisory Committees.

AIR CLEANING

The TAC on Air Cleaning met twice during 1958. Two papers, resulting from the program, *The* ASHAE *Airborne Dust Survey* by Whitby, Algren, Jordan and Annis, and *Evaluation of Air Cleaners for Air Conditioning and Ventilation*, by Whitby, Algren, Jordan and Annis, were published and an additional paper is now being reviewed for presentation in June 1959.

University of Minnesota: For several years the Society has sponsored a research project to study the basic character of particulate matter present in the air and the factors influencing its removal. Nationwide samplings of the air found in industrial cities have been taken and the size, type and distribution of contaminants noted. With this knowledge, test dusts of average particle type and size were developed for use in seeding air

EXPERIMENTAL GAS-FIRED FURNACE USED AT BATTELLE MEMORIAL INSTITUTE IN STUDYING OSCILLATION AS A PART OF THE COOPERATIVE RESEARCH PROJECT WITH THE SOCIETY LABORATORY

under controlled and reproducible laboratory conditions. Apparatus for loading the air with measured quantities of test dust have also been developed as a part of the program. The ultimate goal of the present phase of the Air Cleaning Program is to provide industry with standard test dusts and to design the mechanical equipment necessary to establish an adaptable means of rating the efficiency of air-cleaning equipment.

AIR DISTRIBUTION

The TAC on Air Distribution met at the Semi-Annual Meeting in Minneapolis, Minnesota. One of the items discussed was the need for noise- and pressure-loss studies in turning vanes and take-off fittings. This subject is now being investigated in the Laboratory with the immediate activity being associated with the study of noise conditions in turning vanes.

Mr. Huebner is chairman of a subcommittee reviewing material for THE GUIDE on performance of high-velocity ducting. This work is planned for further discussion at the Philadelphia meeting. Mr. Phillips is serving as chairman of a subcommittee to consider pressure losses in flexible ducts. Methods of testing are being analyzed and checked with manufacturers and users of equipment. When this is completed, a summary of results will be prepared for consideration by the Committee.

Two cooperative research projects are being carried out under the guidance of the TAC.

Case Institute of Technology:—The project is entitled, Model Studies in Air Distribution, with work started on this in the late spring of 1958.

Kansas State College:—The project is entitled, Determination of Heat Losses and Gains from Metal and Non-Metallic Ducts Carrying High-Velocity Air, and work started in late 1958.

COMBUSTION

The TAC on Combustion has had a most active year. Three meetings were held, one on January 28 at the Annual Meeting in Pittsburgh, the second on April 30 at *Battelle Memorial Institute* in Columbus, and the third on June 23 at the Semi-Annual Meeting in Minneapolis. A fourth is planned for the near future.

During this period, the project at *Battelle Memorial Institute* on Oscillations and Pulsations in Oil- and Gas-Fired Domestic Heating Equipment was completed and the results of this work are now being presented to the Society in a series of papers. The first one, covering the results on Multiple-Port Gas Burners, was presented at the Semi-Annual Meeting in Minneapolis and the second on Single-Port Gas Burners, will be presented at the Annual Meeting in January of 1959. Three more papers are forthcoming, one on the Study of Oil-Fired Equipment, and two more covering Suggested Field Cures for Gas- and Oil-Fired Units.

In new projects, this Committee reviewed some 20 subjects and from this group selected 3 as most worthy of further study. A detailed outline of the first phases of these projects is being prepared by a subcommittee for presentation to the Research Committee for approval.

In addition to these activities, this Committee also reviewed and revised Chapter 14, Fuels and Combustion, for THE GUIDE 1959.

Plans are also under way for an invitational meeting to discuss industry problems concerning combustion to aid the committee in setting up long-range programs of activities that would be of greatest interest to the profession.

CONTROL

On May 14, the Research Laboratory was the scene of a Control Conference, planned by a committee under the chairmanship of H. W. Alyea. The committee arranged a program of speakers to present and discuss topics related to the problems of control. A very active discussion, with audience participation, followed the remarks made by the speakers. The Conference before closing went on record that the field of control should become an active part of the research program and that a Technical Advisory Committee should be formed. Following the Conference, the TAC was formed and has been engaged in preliminary discussions throughout the year as to the program that should be

followed. The first formal meeting of the committee is planned to take place at the time of the Annual Meeting in Philadelphia.

EVAPORATIVE COOLING

The Technical Advisory Committee on Evaporative Cooling at its meeting in January 1957, finished discussing the manuscripts its members had prepared for Chapters 34 and 34A of THE GUIDE 1959. Some revisions were made at the meeting, and plans were made so that the final text of the chapters could be prepared. A meeting of the committee is planned for the Annual Meeting in Philadelphia.

HEAT PUMP

The TAC on the Heat Pump has directed extensive effort to the writing of a chapter on Heat Pumps for THE GUIDE 1959. This is an entirely new chapter containing a compilation of the latest information and separate, for the first time from the chapter on Refrigeration. The work, in keeping abreast with the increased use of the heat pump, deals with the history and background, basic cycles, performance characteristics, heat sources, basic components, design selection and heat storage. Authorship is entirely by present and former members of the TAC.

One meeting was held during the year and 2 technical papers, prepared under the auspices of the committee, were presented at the 1958 Annual Meeting.

HEAT TRANSFER THROUGH FENESTRATION

The Committee met in Pittsburgh last January and plans another meeting at the Annual Meeting in Philadelphia in January, 1959.

During the year, the program outlined in Pittsburgh was carried out at the Laboratory. This program has been concerned with measuring the effectiveness of various types of shading devices including exterior awnings, roller shades, interior drapes and between-glass shading. In this connection, it should be noted that a definite technique has been developed and a large number of devices tested and analyzed. Although there remains routine work to round out the knowledge on different types and designs, the basic data in this field have now largely been determined.

During the year, 2 papers were completed, one dealing with metal awnings, the other with roller shades. The work on draperies is nearing completion, and this will be followed by detailed studies on a large number of between-glass shading designs. Following this, it is proposed to prepare a comparative paper on the performance and effectiveness of all shading devices.

At the meeting in January, discussion will be given to planning the subsequent program to be carried out under the auspices of the Committee.

HEATING AND AIR-CONDITIONING LOADS

The TAC met at the time of the 1958 Annual Meeting in Pittsburgh, Pennsylvania. Items of general interest to the Committee were discussed, and reports of the 3 subcommittees (Heating Load, Air-Conditioning Load, and Infiltration) were received.

The activities of the subcommittee of the TAC on Heating Load have been directed toward a revision of Chapter 12 of THE GUIDE. The results of this work should appear in the 1959 edition of THE GUIDE.

Portions of Chapter 13 of THE GUIDE have been reviewed by the Air-Conditioning Loads Subcommittee. This work has not yet been completed, but some new information on internal load sources should appear in the 1959 edition of THE GUIDE.

The first phase of the investigation of infiltration through building entrances has been completed. This work was conducted at the Laboratory and in the field by the Research staff, under the supervision of the Infiltration Loads Subcommittee. Additional work

on infiltration through building entrances equipped with mechanically operated doors and revolving doors is contemplated. It is also planned that summer infiltration rates in large buildings be studied.

Investigation of infiltration around modern types of windows has also been proposed.

HOT WATER AND STEAM HEATING

This Committee has maintained a high level of activity throughout 1958. Under its guidance, new steam flow charts and tables, based on the Moody friction factor were completed at the Laboratory for publication in THE GUIDE 1959. The Laboratory also started a research project to study steam and condensate flow in ferrous and non-ferrous pipe and tubing. Test equipment has been installed and, after preliminary check-out tests on steel pipe, work was begun on non-ferrous pipe. This phase of the project will furnish needed additional factual information regarding condensate formation and flow in copper tubing. The Committee is also giving guidance to 2 cooperative research projects.

University of Illinois: The project on the venting of hot water systems is continuing. This research is studying the effects of boiler-water pressure, location of compression tanks, piping arrangements and chemical treatment of boiler water in relation to gas formation and the points at which gas collects in systems.

Northwestern University: Work was resumed during the year on a study of noise in liquid piping systems. The present study will cover natural sound frequencies of piping spans and resonance conditions in liquid circuits. Continued attention is being given to the problem of sound absorption devices in liquid flow systems.

The Committee held 2 meetings and several subcommittee meetings during 1958. Possible new research in the field of high-temperature water and for a study of the piping and pick-up factors for boilers is being planned.

INDUSTRIAL ENVIRONMENT

Members of the TAC on Industrial Environment have been concerned with the development of useful information on the control of industrial environments to increase human comfort and efficiency and the lessening of air contamination due to industrial processes. Subcommittees have been appointed to investigate the physiological effects of air motion and convective heat; the art of supplying air to workers at industrial operations where physical stress may be caused by adverse environmental conditions; the proper sizing and placing of fans and mechanical equipment with regard to building design and use: industrial air-pollution problems with relation to exhaust systems and air cleaning requirements. The Committee has closely followed the work of T. F. Hatch of the University of Pittsburgh on the design of exhaust hoods for hot industrial processes and the present research there with a study of the problems involved in the use of lateral exhaust ventilation systems.

In addition, members of the committee are preparing new reference information for publication in THE GUIDE and a series of articles on various phases of industrial environmental problems for general publication. The TAC met once in 1958.

INSULATION

This TAC has had an active year with full committee meetings held at the time of the Annual Meeting, January 28, 1958; in a two-day meeting March 25–26 at the Laboratory, and a meeting during the Semi-Annual Meeting, June 23, 1958. In addition to these meetings, several task groups also met. Because a large number of new members were added to the committee this year in connection with the new regulations concern-

ing length of membership on TA committees, the task groups were rearranged with the current organization duty assignments as follows:

Task Group I, Good Practice Recommendations for the Installation of Building Insulation, with G. A. Erickson, chairman, is drafting a proposed chapter for THE GUIDE which is expected to be completed in 1959.

Task Group II, Good Practice Recommendations for the Installation of Industrial Insulation, with J. M. Barnhart, chairman, in conjunction with the *ASRE* Insulation Technical Committee is preparing a chapter for use in THE GUIDE and the *ASRE Data Book*. This should be completed in 1959.

Task Group III, Moisture In Insulated Built-Up Roofs and the Effect of Moisture on Thermal Conductivity, with F. A. Joy, chairman, is studying this subject and is expected to report periodically.

Task Group V, Foundations, with W. J. Binder, chairman, was studying heat loss from slabs. With the completion of former *Task Group IV's* work on *Attic Ventilation*, that group was merged with *Task Group V* to study the heat loss from crawl spaces. *Task Group V* now has the responsibility of investigating heat loss from slab, crawl space and basement foundations. It is expected to offer revisions for THE GUIDE 1960.

Task Group VI, Loose-Fill Insulation Settlement, with E. L. Perrine, chairman, is collaborating with the *ASTM* on this subject.

Task Group VII, Revision of Chapter 9, with C. F. Pridmore, chairman, has prepared major revisions for Chapter 9 which have been approved by the TAC and is appearing in THE GUIDE 1959. This group has under consideration a number of other suggested revisions which it will act on, prior to publication of THE GUIDE 1960.

Task Group VIII, Parallel and Series Heat Flow, with F. A. Joy, chairman, has offered to the TAC a proposed calculation method for this subject. If acted upon favorably, it will appear in THE GUIDE 1960.

Task Group IX, ASA Application Standards for Thermal Insulating Materials, with G. R. Munger, chairman, is at this time acting in an administrative capacity. It will work with *ASA* when *Task Groups I* and *II* have completed their work.

Task Group X, Standardization of Commercial Testing Laboratories. The TAC chairman, G. R. Munger, is working with a committee set up by the American Council of Independent Testing Laboratories to achieve better hot plate test result agreement among commercial testing laboratories.

ODORS

Three meetings of the TAC were held during the year, at the Annual and Semi-Annual Meetings and finally at the Laboratory in Cleveland in September. Work during the year led to completion at the Laboratory, under direction of the TAC, of the research project concerned with development of a method of testing and measuring the odor adsorption and retention characteristics of certain fabrics.

The committee has found by experience that 2 meetings per year are normally sufficient, one in connection with the Annual or Semi-Annual Meetings and a second meeting at the Laboratory in Cleveland.

The paragraphs dealing with odors in THE GUIDE were deemed to be inadequate and a subcommittee was appointed to study the matter and prepare suggestions for inclusion in a future edition of THE GUIDE.

At present the committee is trying to select one or more projects for future Laboratory investigation. A decision on this should be made at or before the meeting to be held in Philadelphia. Subjects under discussion include the following:

1. Study of odor-control methods.
2. Criteria for odor levels in occupied spaces.
3. Continuation program on adsorption and desorption using common complex odors (organic effluvium) under varying temperature and humidity.
4. Study of the *air-conditioned* smell.

The Committee has given consideration to overlapping areas of interest between the TAC on Odors and other TA Committees, especially Air Cleaning, Sorption, Physiological Research and Human Comfort, and Industrial Environment.

PHYSIOLOGICAL RESEARCH AND HUMAN COMFORT

This TAC was formed by combining the former TAC on Physiological Research and the former TAC on Sensations of Comfort into one group. This TAC was activated under date of March 10, 1958.

The programs, as set up under the 2 previous TA Committees, were transferred to this newly organized TAC for information and action.

The first meeting was held in Minneapolis at the Semi-Annual Meeting on Monday, June 23, 1958. Because a quorum was not present agenda items were discussed but no formal actions were taken.

SOLAR CALORIMETER AT LABORATORY SHOWING TEST ARRANGEMENTS FOR CONDUCTING TEST ON INSIDE DRAPES. TEST DRAPES ARE HANGING IN WINDOW OPENING

The only direct action taken by the members of the TAC by letter ballot was a recommendation to the Guide Publication Committee to eliminate Fig. 1, p. 115, Chapter VI of THE GUIDE 1958, and to modify the wording of this chapter wherever reference to this figure appears.

The Laboratory was told to proceed on the first phase of the program, as approved by the former TAC on Sensations of Comfort. Phase I of this program is the re-evaluation of the Comfort Chart and a paper in connection with this project has now been prepared for presentation at the Topical Session on Comfort scheduled at the Annual Meeting in Philadelphia, January 26, 1959.

At the Semi-Annual Meeting in Minneapolis, representatives of a commercial organization discussed their experimental work in connection with air ionization and offered to assist the Laboratory in further studying the question of ionization with relation to human comfort by loaning the Laboratory their equipment and advising them on certain techniques which they have developed.

On September 16, a letter was sent to all members of the TAC asking them to return a ballot either in favor or against the preliminary study of ionization in connection with human comfort with the idea that this work could be done at the same time as the evaluation of the Comfort Chart was being made. A majority of the committee favored this action. However, details of the program have not yet been worked out and this topic will be considered in more detail at the next meeting of the committee.

PLANT AND ANIMAL HUSBANDRY

The TAC on Plant and Animal Husbandry met December 18 in Chicago. The time of the meeting was planned to coincide with the Annual Meeting of the *American Society of Agricultural Engineers*. Throughout the year the committee has been actively working on a survey of environmental conditions, particularly in relation to domestic animals. A number of reports have already been prepared, and one important topic for consideration was how best to summarize the reports for general publication and later for a chapter in THE GUIDE.

SOLAR ENERGY UTILIZATION

During the past year, the Technical Advisory Committee on Solar Energy Utilization completed the article, *Solar Energy Utilisation for Heating, Cooling, Distillation and Drying* (published in ASHAE TRANSACTIONS, Vol. 64, 1958, p. 517) which was presented at the 1958 Semi-Annual Meeting of the Society. This was the culmination of a project to determine those areas of solar energy utilization in which research activities are most needed and to pinpoint the status of the knowledge up to the present time.

Since the completion of this activity, the Committee has been exploring additional potential projects which might be possible and desirable to prosecute despite the geographical diffusion of the committee personnel.

SORPTION

The TAC on Sorption held 2 meetings during the year, 1 at the Annual Meeting and 1 at the Semi-Annual Meeting. At the first meeting, discussion revealed that the scope in use did not fully cover the activities of the committee and so a revised scope was prepared which was later approved by the Committee on Research. The new scope is as follows:

Sorption of moisture by materials, especially as it affects latent heat loads and dehumidification, and the study of materials and equipment for moisture control by sorption and related problems.

The previously prepared code for testing sorption dehumidification has not yet been acted upon by the Standards Committee of the Society.

One of the activities of the TAC was to completely rewrite the chapter on Dehumidification by Sorbents for the 1959 issue of THE GUIDE. The committee has been closely following research at the University of Toledo on liquid sorption equipment and at The Pennsylvania State University on solid sorption equipment. Preliminary results from both of these researches indicate that air purification resulting in sterilization can be accomplished with both types of sorption equipment. The work at these 2 institutions is closely followed by 2 subcommittees which have been appointed, and the main committee is working on co-ordinating the activities of these 2 research projects. Plans are under way for setting up additional research projects, one of which now under discussion relates to the possibility of removing odors by liquid and solid sorbents.

SOUND AND VIBRATION CONTROL

The TAC on Sound and Vibration Control has been very active throughout 1958 in assisting the Laboratory staff in carrying out the Sound Program for the Bureau of Ships. This program for which extensive equipment has been installed at the Labora-

SETUP IN RESEARCH LABORATORY TO STUDY SOUND GENERATION AND ATTENUATION IN DUCT ELBOWS. AT RIGHT ARE DUCTS ARRANGED TO TURN AIR THROUGH SUCCESSIVE 90 DEG ELLS. BELOW IS VIEW OF RECORDING STATION AT TERMINATION OF DUCT RUN

tory, was initiated early in 1958 when studies to determine the sound-generating characteristics of various duct system components were begun. Navy-type elbows having turning vanes were tested and their sound-generating characteristics determined. This first phase of the project is now virtually complete and it is planned to continue with a study of the sound-attenuating characteristics of the same components.

In addition, the TAC is cooperating closely with *ASRE* in a joint effort to establish Standards on Equipment Sound Testing. The Committee is promoting the development of a standard sound source for use in the calibrating rooms where equipment sound

levels are to be measured. Several different types of standard sources have been submitted and studied in the course of this work.

The Committee is directing its efforts through 5 subcommittees. From the joint activities of these groups, the Society is represented in a variety of sound activities. A constant investigation of all possible areas of research in sound is also made. The TAC met twice during 1958 and there were several subcommittee meetings. Several members of the TAC also served on the Joint *ASRE*-ASHAE Committee on Equipment Sound Testing.

THERMAL CIRCUITS

Two meetings were held this year; one at the Semi-Annual Meeting in Minneapolis in June and the other at the Division of Building Research, *National Research Council*, Ottawa, Canada, in October 1958.

University of California at Los Angeles: The continuing cooperative project at this university produced a paper, *Cooling Load From Pretabulated Impedances*, by Harry Buchberg, which was presented at the Minneapolis Meeting. Another paper has been reviewed and it is expected that it will be presented at the Semi-Annual Meeting in 1959.

Approval has been obtained for continuation of the UCLA cooperative project as follows:

1. Research leading to evaluation of the importance of including the inside radiation exchange network in load determinations.
2. A preliminary design feasibility study for a low-cost special-purpose computer especially suited for an integrated design approach to air-conditioning problems.
3. Research leading to a study of analogue techniques applied to the utilization of solar energy with flat-plate, glass-covered, collectors in co-sponsorship with the TAC on Solar Energy Utilization.

WEATHER DATA

One meeting of the TAC on Weather Data was held in 1958 at the time of the Annual Meeting.

The group has been active in gathering weather data particularly for publication in THE GUIDE 1959. New weather stations are included and wet-bulb temperatures and wind-speed data are given for an increased number of stations. H. C. S. Thom presented for review his work on *Revised Summer Outside Design Temperatures*. The months included are June, July, August and September. Both dry-bulb and wet-bulb temperatures will be given and the suggested design temperatures will be figured on the TAC 2½-percent basis which fixes the risk of exceeding the temperatures listed to 3 days in any one season. One paper, *Local Climatic Weather Data*, by W. L. Holladay, was presented at the Annual Meeting in Pittsburgh in 1958.

On completion of the Report of the Committee on Research, Vice President Hess stated that the next item on the program was the Report of the Inspectors of Election. He recognized H. M. Patrick, Bernardsville, N. J., who presented the report as follows:

REPORT OF INSPECTORS OF ELECTION

Ballot for Officers, 1959	Total
President, Arthur J. Hess, Los Angeles, Calif.	3111
First Vice President, Walter A. Grant, Syracuse, N. Y.	3108
Second Vice President, John Everetts, Jr., Philadelphia, Pa.	3104
Treasurer, John H. Fox, Toronto, Ont., Canada	3104

Members of Council (three-year term)

Wm. J. Collins, Jr., Oklahoma City, Okla.	3102
Harold A. Lockhart, Morton Grove, Ill.	3097
George W. F. Myers, St. Louis, Mo.	3095
Rolland S. Stover, Marshalltown, Ia.	3097

Committee on Research (three-year term)

Albert Giannini, New York	3097
Richard C. Jordan, Minneapolis	3097
Maurice Nelles, Chicago	3093
Herbert B. Nottage, Encino, Calif.	3093
Paul H. Yeomans, Philadelphia	3094
TOTAL BALLOTS RECEIVED	3243
TOTAL LEGAL BALLOTS	3155
INVALID BALLOTS	88

Scattering votes: Officers 9; Members of Council 6; and Committee on Research 14.

By-Law Amendments

		Total Yes	Total No
1. **Article IV**	*Section* 6 (*a*) Investment of Funds	2780	127
2. **Article VII**	*Section* 3 (*e*) F. Paul Anderson Committee	2890	11
3. **Article VII**	*Section* 3 (*f*) Chapters Regional Committee	2892	7
4. **Article VIII**	*Section* 3 Nominating Committee	2864	33
5. **Article IX**	*Section* 3 Prerequisites	2818	40

Respectfully submitted,

HORACE M. PATRICK, *Chairman*
H. BERKELEY HEDGES
J. K. THORNTON

Four papers had been programmed for presentation at this session, one of which was presented by title only. Each of these papers was successively presented and discussed, after which Vice President Hess adjourned the session at 11:40 a.m.

THIRD SESSION, TUESDAY, JANUARY 27, 9:30 A.M.

This session was organized in the form of a Symposium on Hydronics. It was presided over by Second Vice Pres. Walter A. Grant, Syracuse, N. Y., and P. N. Vinther, Dallas, Tex., was moderator. Four papers had been assembled for this session. One of these entitled, Design of Water Piping Systems, by B. L. Sturdevant, Muscatine, Ia., is printed substantially in full in the February 1959 ASHAE JOURNAL, beginning on p. 13. The second paper, Noise and Vibration in Water Piping Systems, by W. L. Rogers, Evanston, Ill., can be found in the ASHRAE JOURNAL, March 1959, beginning on p. 83. A third paper entitled, Selection of Pumping Equipment, by G. F. Carlson, Morton Grove, Ill., was published in the ASHAE JOURNAL, February 1959, beginning on p. 3. The fourth paper in this Symposium is entitled, Refrigerant Piping, by J. R. Chamberlain, York, Pa.

These 4 Symposium papers are not printed in the TRANSACTIONS, but are available from the Society in full, together with the discussions which took place at the session, in a separately printed Bulletin.

FOURTH SESSION, WEDNESDAY, JANUARY 28, 9:30 A.M.

Treasurer Fox presided at the fourth session at which 5 papers were presented and discussed, one of them being made available by title only.

FIFTH SESSION, WEDNESDAY, JANUARY 28, 9:30 A.M.

This session took the form of a Symposium on Heat Pump Performance and was called to order by President Queer. He turned the session over to G. B. Priester, Baltimore, Md., who was the moderator for the Symposium. Five papers had been assembled for this session and were presented by their authors and were discussed from the floor. Of these, the paper, Operation of Large Air Source Central Heat Pump Systems, by E. R. Ambrose, New York, N. Y., can be found in the ASHAE JOURNAL, February 1959, beginning on p. 21, and the one entitled, Performance of Package Air Source Heat Pump Units, by W. A. Spofford, Tyler, Tex., can be found in the ASHRAE JOURNAL, April 1959, beginning on p. 59.

The 5 papers in this Symposium are not printed in the TRANSACTIONS, but their full text is available from the Society, together with the discussions which took place at the session, in the form of a separately printed Bulletin.

SIXTH SESSION, WEDNESDAY, JANUARY 28, 2:00 P.M.

This session was opened by Vice President Hess, and 3 papers as listed on the Program were presented and discussed.

SEVENTH SESSION, WEDNESDAY, JANUARY 28, 2:00 P.M.

This session was organized as a Symposium on Corrosion and Water Treatment and opened with President Queer presiding. The session program had been arranged by W. G. Hole, Montreal, Que., Canada, who acted as moderator. Five papers had been prepared and were successively presented and discussed. One of these papers entitled, Water Treatment to Inhibit Corrosion, by F. S. Hodgdon, Long Island City, N. Y., can be found in the ASHAE JOURNAL, February 1959, beginning on p. 29. The paper entitled, Thermal Insulation and Corrosion, by C. E. Ernst and G. P. Walker, New York, N. Y., is printed in the ASHAE February JOURNAL, 1959, beginning on p. 26. The third of the 5 papers, Hot Water and Anti-Freeze Systems, by L. D. Polderman, New York, N. Y., is also published in the ASHAE JOURNAL, February 1959, and begins on p. 9.

None of these Symposium papers are printed in TRANSACTIONS but they are available from the Society in the form of a separately printed Bulletin containing the text of all 5 of the papers and the discussions from the floor at the session.

Following the completion of the Symposium Session, President Queer stated that the next order of business was the Report of the Committee on Resolutions by its chairman, S. F. Gilman, Syracuse, N. Y., as follows:

RESOLUTIONS

WHEREAS, the 65th Annual Meeting of the SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS and the 14th International Heating and Air-Conditioning Exposition are now drawing to a close, and

WHEREAS, this meeting was held in Philadelphia where all varieties of weather were experienced in a few short days, and

WHEREAS, the Philadelphia Chapter and the Committee on Arrangements under the direction of E. K. Wagner, general chairman, have ably provided for our comforts, entertainment and relaxation, and

WHEREAS, the technical life of the Society has been advanced through the presentation and discussion of a series of well-prepared papers and symposia, and

WHEREAS, the 14th International Heating and Air-Conditioning Exposition has broken all records for quality of exhibits, size of attendance; and

BE IT RESOLVED, that our appreciation and sincere thanks be extended to:

The Philadelphia Chapter, its officers and members who have made this meeting possible,

E. K. Wagner, general chairman; F. H. Buzzard, vice-chairman; and to J. H. Hucker and R. P. Schoenijahn, honorary chairmen,

Mr. and Mrs. R. F. Buzzard and their committee of ladies who so outstandingly provided for the welfare and entertainment of our visiting ladies,

Dr. James Creese of Drexel Institute of Technology who so ably pinch hit for the Mayor at our Welcome Luncheon,

The authors and discussers of the technical papers and symposia for their educational contributions,

Dr. Eric A. Walker whose speech at the annual banquet we look forward to with great anticipation,

Professor Merl Baker for his technical contributions to our profession and who received the Life Members' Award,

The outstanding student, James Y. Payton, recipient of the ASHAE-Homer Addams Award,

M. F. Blankin, B. H. Jennings, R. C. Jordan, B. H. Spurlock, Jr., and Reg. F. Taylor, who were awarded the grade of Fellow,

E. K. Stevens and the International Exposition Company for the excellent presentation of the products of our industry,

The headquarters staff for its continued efficient handling of all the myriad details involved in our meetings,

Pres. E. R. Queer for an outstanding job during this extremely important year in the Society's history, and to his fellow Officers and Council Members who have so ably assisted him,

All those whose continued efforts made possible a marriage and a life with *ASRE* at our ripe old age of 65.

Respectfully submitted,

RESOLUTIONS COMMITTEE

S. F. GILMAN, *Chairman*, Syracuse, N. Y.
C. B. GAMBLE, New Orleans, La.
W. J. OLVANY, New York, N. Y.

On completion of the reading of the resolutions, and on motion, duly made and seconded, President Queer declared their adoption.

President Queer then asked if there were any items of unfinished or new business to come before the session, and since there appeared to be none, he declared the session adjourned at 4:30 p.m.

PROGRAM—65th ANNUAL MEETING ASHAE

Bellevue Stratford Hotel, Philadelphia, Pa. **January 26–29, 1959**

Saturday, January 24

10:00 a.m. Joint Executive Committees (*Blue Room*)
2:00 p.m. Research Executive Committee (*Gold Room*), E. F. Snyder, Jr., *Chairman*
2:30 p.m. Joint Membership Committee (*Room 108*)
3:30 p.m. ASHAE Finance Committee (*Room 105*)
4:00 p.m. ASHAE Program and Papers Committee, John Everetts, Jr., *Chairman* (*Room 107*)

Sunday, January 25

7:30 a.m. ASHAE Executive Committee (*Gold Room*)
9:30 a.m. Council Meeting (*Crystal Room*)
10:00 a.m. Committee on Research (*Green Room*), E. F. Snyder, Jr., *Chairman*
10:00 a.m. ASHAE Admission & Advancement Committee, (*Room 106*)
2:00 p.m. Joint Regional Directors Meeting (*Crystal Room*)
3:00 p.m. Welcome—Pennsylvania Dutch Koffee Klatch, (*Clover and Red Room*)
8:30 p.m. Movies (*Clover Room*)

Monday, January 26

8:30 a.m. REGISTRATION (*Ballroom Floor*)
9:00 a.m. FIRST SESSION (*Clover and Red Rooms*)
Call to Order—Pres. E. R. Queer
Report of President—E. R. Queer
Report of Treasurer—John H. Fox
Report of Council—A. V. Hutchinson
Amendments to the By-Laws
TOPICAL SESSION ON COMFORT
Charles S. Leopold, *Moderator*
Researches on Heating and Ventilation in Relation to Human Comfort, by Thomas Bedford, London, England, presented by Dr. Bedford
Metabolic Patterns in Stressful Environments, by Nello Pace, Berkeley, Calif., presented by Dr. Pace
Environment Reactions in the 80 to 105 F Zone, by B. H. Jennings and Baruch Givoni, Cleveland, Ohio, presented by Mr. Jennings.
9:30 a.m. Ladies Bus Tour of Old Philadelphia (*Assemble Viennese Room, Lobby Floor*) Buses leave from Bellevue Stratford
10:00 a.m. Joint Admission & Advancement Committees (*Room 106*)
12:00 noon EXPOSITION—Opening of 14th International Heating and Air-Conditioning Exposition—Convention Hall
12:15 p.m. WELCOME LUNCHEON (*Ballroom*)
Toastmaster: E. K. Wagner, *General Chairman*
Invocation: P. H. Yeomans
Speaker: Dr. James Creese, President, Drexel Institute of Technology
Subject: Independence Hall and the Kremlin

Presentation of Certificates to Fellows: M. F. Blankin, Philadelphia, Pa.; B. H. Jennings, Cleveland, Ohio; R. C. Jordan, Minneapolis, Minn.; B. H. Spurlock, Jr., Boulder, Colo.; and Reg. F. Taylor, Houston, Tex.

Presentation of ASHAE-Homer Addams Award to James Y. Payton, Los Angeles, Calif.

2:00 p.m. Ladies' Bus Tour of Old Philadelphia (*Assemble Viennese Room, Lobby Floor*) Buses leave from Bellevue Stratford

2:00 p.m. TAC on Air Cleaning (*Blue Room*), W. C. L. Hemeon, *Chairman*

2:00 p.m. TAC on Air Distribution (*Green Room*), P. A. Argentieri, *Chairman*

2:00 p.m. TAC on Heat Transfer Through Fenestration (*Room 105*), D. D'Eustachio, *Chairman*

2:00 p.m. TAC on Hot Water and Steam Heating (*Room 106*), A. O. Roche, Jr., *Chairman*

2:00 p.m. TAC on Odors (*Room 107*), H. L. Barnebey, *Chairman*

2:30 p.m. Joint Publication Committees (*Secretary's Suite*)

2:30 p.m. Joint Building Committees (*Gold Room*)

7:30 p.m. Get-Together Party (*Grand Ballroom—Sheraton Hotel*)

Social Hour—Buffet Dinner—Entertainment

Dancing—Howard Lanin Orchestra

Tuesday, January 27

9:00 a.m. REGISTRATION (*Ballroom Floor*)

9:30 a.m. SECOND SESSION (*Clover and Red Rooms*)

Call to Order—1st Vice Pres. A. J. Hess

Report of Committee on Research—E. F. Snyder, Jr., *Chairman*

Report of Inspectors of Election

Supply Outlet Locations for Basement Heating, by J. R. Wright and D. R. Bahnfleth, Urbana, Ill., presented by Mr. Wright

Pulsations in Single-Port Gas-Fired Residential Heating Equipment, by C. F. Speich and A. A. Putnam, Columbus, Ohio, presented by Mr. Speich

Low Frequency Combustion Noise in Oil-Burning Equipment, by G. J. Sanders and W. E. Lawrie, Chicago, Ill., presented by Mr. Sanders

Omni-Directional Low-Speed Hot-Wire Anemometer, by J. F. Kemp, Pretoria, South Africa (by title only)

9:30 a.m. THIRD SESSION (*Ballroom*)

Call to Order—2nd Vice Pres. Walter A. Grant

SYMPOSIUM ON HYDRONICS

P. N. Vinther, *Moderator*

Design of Water Piping Systems, by B. L. Sturdevant

Noise and Vibration in Water Piping Systems, by W. L. Rogers

Selection of Pumping Equipment, by G. F. Carlson

Refrigerant Piping, by J. R. Chamberlain

Report of Inspectors of Election

9:45 a.m. Ladies' Bus Tour of Historical Mansions, (*Assemble Viennese Room, Lobby Floor*)

Luncheon (*Madison Room, Presidential Apartments*) Buses leave from Bellevue Stratford

12:00 noon EXPOSITION—14th International Heating and Air-Conditioning Exposition —Convention Hall (Closes 10 p.m.)
12:15 p.m. Life Members' Luncheon (*Crystal Room*)
1:30 p.m. Joint Standards Committees (*Secretary's Suite*)
2:00 p.m. Conference of Chapter Editors and Publicity Chairmen (*Blue Room*)
2:00 p.m. TAC on Heat Pump (*Gold Room*), P. R. Achenbach, *Chairman*
2:00 p.m. TAC on Heating and Air Conditioning Loads (*Room 105*), H. T. Gilkey, *Chairman*
2:00 p.m. TAC on Insulation (*Room 106*), G. R. Munger, *Chairman*
2:00 p.m. TAC on Physiological Research and Human Comfort (*Green Room*), John Everetts, Jr., *Chairman*
2:00 p.m. TAC on Sound and Vibration Control (*Room 107*), A. F. Hubbard, *Chairman*
2:00 p.m. TAC on Thermal Circuits (*Room 108*), S. F. Gilman, *Chairman*
7:00 p.m. Past Presidents' Dinner (*Union League, 140 S. Broad Street*)
8:00 p.m. TAC on Combustion (*Room 105*), M. W. McRae, *Chairman*
8:00 p.m. TAC on Control (*Room 106*), H. W. Alyea, *Chairman*
8:00 p.m. TAC on Industrial Environment (*Room 107*), Arthur Nutting, *Chairman*
8:00 p.m. TAC on Solar Energy Utilization (*Room 108*), R. C. Jordan, *Chairman*
8:00 p.m. TAC on Sorption (*Pink Room*), G. C. F. Asker, *Chairman*
8:00 p.m. TAC on Weather Data (*Green Room*), W. L. Holladay, *Chairman*

Wednesday, January 28

9:00 a.m. REGISTRATION (*Ballroom Floor*)
9:30 a.m. FOURTH SESSION (*Burgundy Room, Lobby Floor*)

Call to Order—Treas. J. H. Fox

Frost Formation and Heat Transfer on a Cylinder Surface in Humid Air Cross Flow, Parts I and II, by P. M. Chung and A. B. Algren, Minneapolis, Minn., presented by Mr. Chung.

Downward Heat Transfer Through a Joist Space, by P. M. Chung and C. E. Lund, Minneapolis, Minn. (by title only).

Annual Energy Use for Electric Space Heating, by R. L. Boyd, Pittsburgh, Pa., presented by Mr. Boyd.

Heating by Hypocaust, by Edwin D. Thatcher, New York, N. Y., presented by Mr. Thatcher.

9:30 a.m. FIFTH SESSION (*Ballroom*)

Call to Order—E. R. Queer

SYMPOSIUM ON HEAT PUMP PERFORMANCE

G. B. Priester, *Moderator*

Performance of Water Source Heat Pump for Office Building in Southwest, by F. L. McFadden, Jr.

Performance of Water Source Heat Pumps in the Pacific Northwest, by J. Donald Kroeker

Performance of Package Air Source Heat Pump Units, by W. A. Spofford

Operation of Large Air Source Central Heat Pump Systems, by E. R. Ambrose

Performance of an Air Source Compound Heat Pump, by A. D. Spillman

12:00 noon EXPOSITION—14th International Heating and Air Conditioning Exposition —Convention Hall (Closes 10 p.m.)

12:15 p.m. Authors' Luncheon (*Blue Room*)

12:30 p.m. Ladies' Luncheon and Fashion Show by John Wanamaker

2:00 p.m. SIXTH SESSION (*Burgundy Room, Lobby Floor*)

Call to Order—1st Vice Pres. A. J. Hess

Probe for Thermal Conductivity Measurement of Dry and Moist Materials, by W. Woodside, Ottawa, Canada, presented by Mr. Woodside.

Heat Gain through Windows Shaded by Metal Awnings, by Necati Ozisik and L. F. Schutrum, Cleveland, Ohio, presented by Dr. Ozisik.

A New Method for Calculating Radiant Exchanges, by Benjamin Gebhart, Ithaca, N. Y., presented by Mr. Gebhart.

2:00 p.m. SEVENTH SESSION (*Ballroom*)

Call to Order—Pres. E. R. Queer

SYMPOSIUM ON CORROSION AND WATER TREATMENT

W. G. Hole, *Moderator*

Water Treatment to Inhibit Corrosion, by F. S. Hodgdon

Thermal Insulation and Corrosion, by C. E. Ernst and G. P. Walker

Hot Water and Anti-Freeze Systems, by L. D. Polderman

Taking the Mystery Out of Water Treatment, by Mrs. Olivia Burkey

Specification Problems in Water Treatment by W. L. Andrews

Report of Resolutions Committee

Unfinished Business—New Business

Adjournment

6:00 p.m. Social Hour (*Clover Room*)

7:15 p.m. 65th ANNUAL BANQUET (*Ballroom*)

Toastmaster: Albert J. Nesbitt

Invocation: William D. Powell, General Secretary, Greater Philadelphia Council of Churches

Speaker: Dr. Eric A. Walker, president, The Pennsylvania State University

Subject: The Morning After for Education

Introduction and Installation of 1959 Officers

Presentation of F. Paul Anderson Medal to C. P. Yaglou by Pres. Queer

Presentation of Past President's Award to E. R. Queer by P. B. Gordon

Dancing (*Rose Garden, 18th Floor*) from 10:00 p.m.
Howard Lanin Orchestra

Thursday, January 29

9:00 a.m. Council Meeting

9:00 a.m. Guide Committee (*Room 108*)

9:30 a.m. Joint Meeting Committee on Research ASHAE-*ASRE* (*Blue Room*)

10:00 a.m. Auf Wiedersehen Coffee Hour (*Clover Room*)

10:30 a.m. Joint Meeting Executive Committees ASHAE-*ASRE* (*Burgundy Room*)

12:00 noon EXPOSITION—14th International Heating and Air-Conditioning Exposition —Convention Hall (Closes at 10:00 p.m.)

2:30 p.m. Joint Meeting Program and Papers Committees ASHAE-*ASRE*

Friday, January 30

9:00 a.m. Joint Finance Committees (*Blue Room*)

COMMITTEE ON ARRANGEMENTS

E. K. WAGNER, *General Chairman*

F. H. BUZZARD, *Vice-Chairman*

J. H. HUCKER, R. P. SCHOENIJAHN, *Honorary Chairmen*

Banquet—J. J. Hucker, *Chairman*, P. R. Anderson, J. C. Benson, K. E. Clisby, L. J. Dorsey, J. R. Ellstrom, J. K. Henderson, H. N. Jacobs, H. R. James, S. J. Kowalski, Hugh Mahaffy, M. J. Wind, F. J. Zellner.

Entertainment—H. B. Prewitt, *Chairman*, R. H. Ackermann, E. I. Curley, E. H. Dafter, F. R. Kelble, T. S. Noble, C. W. Proctor, Jr., R. J. Sigel, R. H. Smiles, G. E. Tuckerman, D. A. Wicks, E. I. Wildman.

Exposition—C. J. Lubking, *Chairman*, T. L. Arnold, V. R. Bitzer, I. G. Brennan, Warren Brewer, C. F. Dietz, H. W. Doerrfuss, C. J. Kern, H. K. King, A. E. Kriebel, J. F. Lowrey, H. T. Makin, Jr., H. H. Mohrfeld, C. A. Rhea, H. N. Teuber.

Finance—P. H. Yeomans, *Chairman*, A. C. Caldwell, W. R. Funk, A. J. Hall, Jr., J. E. Hutchison, E. C. Lambert, F. F. Pfeiffer, G. H. Johnson.

Ladies—R. F. Buzzard, *Chairman*, Mrs. R. F. Buzzard, *Co-Chairman*, Isadore Director, Mmes. R. H. Ackermann, P. Russell Anderson, R. S. Arnold, J. C. Benson, M. F. Blankin, F. H. Buzzard, P. A. Cavanagh, E. H. Dafter, C. F. Dietz, John Everetts, Jr., C. J. Forve, J. H. Hucker, J. J. Hucker, C. J. Lubking, Ludwig Mack, D. S. Plewes, H. B. Prewitt, A. M. Robertson, R. P. Schoenijahn, R. J. Sigel, E. J. Stern, E. K. Wagner, M. J. Wind, P. H. Yeomans, Isadore Director, A. H. Koch and R. M. Sharp.

Publicity—Ludwig Mack, *Chairman*, G. P. Alexander, T. J. Atkins, L. A. Dickson, C. J. Moll, Morris Sheffler.

Reception—P. A. Cavanagh, *Chairman*, R. S. Arnold, R. J. Augusterfer, Lucien Buck, G. F. Bertrand, C. E. Brock, Jr., C. B. Eastman, Edwin Elliott, H. H. Erickson, J. P. Hartmann, H. B. Hedges, R. F. Hunger, M. G. Kershaw, C. S. Leopold, N. W. MacNichol, W. W. Malloy, S. W. Miller, Jr., R. S. Mizener, A. G. Mutimer, H. M. Parent, Karl Rugart, W. J. Searle, B. S. Urmston, K. W. Will, C. F. Wiltberger.

Sessions—A. M. Robertson, *Chairman*, R. W. Cromie, R. G. Lau, Jr., R. M. MacLean, E. G. Ramp, E. J. Stern, J. A. Zimmerman.

Transportation—D. S. Plewes, *Chairman*, R. L. Bitzer, J. E. Copeland, L. A. Davidson, M. M. Garber, J. M. Johnston, H. V. Mount, Jr., M. J. Murray, G. W. Powell, 3rd, Edward Semel, D. J. Smith, R. B. Twining, J. A. Youtz.

Welcome Luncheon—C. J. Forve, *Chairman*, W. P. Culbert, E. J. Ewald, R. M. Gillett, T. F. Kearney, S. H. Manheimer, J. W. McElgin, Stanley Stake, Leo Sudhaus.

Auf Wiedersehen Coffee Hour—P. Russell Anderson, *Chairman*; the Messrs. and Mmes. R. S. Arnold, J. C. Benson, E. H. Dafter, John Everetts, Jr., E. J. Stern, E. K. Wagner.

1653

No. 1653

RESEARCH ON HEATING AND VENTILATION IN RELATION TO HUMAN COMFORT

By THOMAS BEDFORD*, LONDON, ENGLAND

THAT THE WARMTH of man's surroundings has a profound influence on his well-being is no new idea. Before the Christian era Hippocrates wrote of the effect of climate on the physique, intelligence and temper of peoples. Modern climatologists[1, 2] have studied the effects of climate on civilization. Markham holds that civilization depends to a great extent upon the control of the indoor climate in a good natural climate.

In the past half century many investigations of the effects of the thermal environment on comfort and on efficiency and general well-being have been made, and on the whole it seems reasonable to conclude that the conditions desirable for comfort are also those most conducive to efficiency.

In discussing the subject of comfort the author proposes to draw largely on the work of his associates and himself.

THERMAL MEASUREMENTS

First, reference is made to measurements of the thermal environment. Measurements of air temperature and humidity present no difficulty and call for no comment

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This paper and Nos. 1654 and 1655 were presented at a Topical Session on Comfort. At the opening of that Session, and in the course of introducing the author, Dr. Bedford, the moderator, C. S. Leopold, Philadelphia, Pa., made the following statement:

"There is a considerable difference in English and North American practice as regards maintained temperatures. It has been my privilege to discuss this problem with Dr. Bedford on several occasions. I think you will find, as I have found, that the differences in practice can be reasonably explained.

"There is no widely accepted definition of the word 'comfort.' THE GUIDE makes no attempt to define it. For American practice I have chosen to define optimum comfort as a condition the majority of people would prefer if they did not have to pay for obtaining the result and did not know a test was in progress. This definition, I assure you, is not official.

"Some investigators choose to include stimulation and health in considerations of comfort. This is certainly an equally admissable approach as long as we know which type of comfort we are discussing."

* Director of the Medical Research Council's Environmental Hygiene Research Unit, London School of Hygiene and Tropical Medicine.

[1] Exponent numerals refer to References.

Presented at the 65th Annual meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, January 1959.

Air speed: For the measurement of air speeds the silvered kata thermometer has been relied greatly upon, but this gives only the average speed over a period of 2 or 3 min. When concerned with the variability of speed a hot-wire anemometer consisting of 3 wires in mutually perpendicular planes was used. A difficulty with many hot-wire anemometers, however, is that at low air speeds the natural convection due to the heated wire may mask the effects of the general air movement.

FIG. 1. . . .THE IONIZATION ANEMOMETER

A few years ago, with the object of overcoming this difficulty, Dr. Lovelock[3] devised an ionization anemometer (Fig. 1) which is omnidirectional. In the original model the ionizing source was a ½-in. sphere plated with polonium. When it was later shown that particles of polonium could be dispersed through disintegration, and that they were highly toxic, it was necessary to use another source. In the modified instrument[4] the ionizing electrode is covered with radium-gold foil (radium sandwiched between 2 films of gold, backed on one side with silver foil). Thus radon is occluded and the polonium which is eventually produced is prevented from escaping. Surrounding the active sphere is a collecting cage of 3 mutually perpen-

dicular rings of 10-cm diameter, and this is surrounded by an earthed screen. The ion current is fed to an amplifier and the anode current is calibrated in terms of air speed. The instrument can be used with an oscilloscope and camera or with a high-speed recorder, and will give full response with frequencies up to 2 cps.

Mean Radiant Temperature: The mean radiant temperature (MRT) has been measured either by scanning the surroundings with a radiometer (a Moll thermopile), or it has been computed from readings of the globe (6-in. diameter), the air temperature and the air speed[5].

The equation for the globe thermometer is

$$(t_w + 460)^4 \times 10^{-9} = (t_g + 460)^4 \times 10^{-9} + 0.103 \sqrt{V}\,(t_g - t_a)$$

where

t_w = mean radiant temperature, Fahrenheit degrees.

t_g = globe thermometer temperature, Fahrenheit degrees.

t_a = air temperature, Fahrenheit degrees.

V = air speed, feet per minute.

Either of these methods gives the MRT with reference to a point or a small sphere. The value so obtained may not represent accurately the radiation load on a standing man if the radiation is strongly directional.

At the Royal Naval Tropical Research Unit at Singapore estimates of the MRT made from globe thermometers were compared with observations made on *metal men*. These last were rather crude effigies of men, one having a surface of polished metal and the other a black surface, so that the radiation exchanges of these roughly human shapes could be determined. The tests were made in a wind tunnel, the surfaces of which were heated, but where the radiation was by no means uniformly distributed. The MRT was maintained at or above the air temperature and at times the difference was as much as 20 F deg. Globe thermometers were placed 2 ft, 3 ft-9 in. and 5 ft-6 in. above the floor. On occasions the MRT's estimated from the globes differed from the MRT ascertained from the metal men, but when the readings of the 3 globes were averaged the MRT agreed with that obtained from the metal men to within 1 F deg.

It is well to remember that in an occupied room the MRT is affected by the presence of the occupants. The effect was noted 30 years ago by Yaglou and Drinker[6], and later the author made observations[7] in factories (1) when the operatives were at work immediately before or after the dinner interval, and (2) just after the occupants had left or before they re-entered. The increase in MRT due to the occupants varied from 1 F deg in a sparsely occupied room to 4 F deg where people sat fairly close together at rows of benches. The effect would be greater still in a crowded place of assembly.

INDEXES OF WARMTH

Comparison of Indexes: The scale of warmth most widely known and used is effective temperature, but other scales have also been used in Great Britain. Equivalent temperature allows for the temperature and speed of the air and for radiation, but it ignores humidity and is therefore of little value at temperatures

much above 70 F. For ordinary room conditions Leonard Hill advocated the *cooling power* as denoted by the dry kata thermometer. Vernon suggested that the simple reading of the globe thermometer might be a good index, since it was affected by radiation.

Many years ago the author compared these indexes in a study of the comfort of people performing light industrial operations in British factories during the heating season[7]. The comfort votes recorded were correlated with the various indexes of warmth (Table 1). The number of observations in the main series was 2571.

Some of the workers were habituated to conditions that were warm by British standards and others to cool conditions, so it was not surprising that the correla-

TABLE 1. . . .CORRELATION OF COMFORT VOTES WITH INDEXES OF WARMTH

COMFORT VOTES CORRELATED WITH	CORRELATION COEFFICIENT (*r*)
Equivalent temperature	0.52 ± 0.01
Globe thermometer reading	0.51 ± 0.01
Dry-bulb temperature	0.48 ± 0.01
Effective temperature	0.48 ± 0.01
Dry kata cooling power	--0.43 ± 0.01

tions were not very high. It was interesting that the 2 indexes that made reasonable allowance for radiation gave the highest correlations, while effective temperature was no better than the simple air temperature. At the air temperatures encountered (mean 64.4 F, s.d.* 3.7 F deg) a considerable change in humidity had very little effect on sensations of warmth, and the air speed (mean 31 fpm, s.d. 15 fpm) did not vary greatly. When later an adjustment was made to the effective temperatures to make an allowance for radiant heat the correlation with comfort votes was raised to the level of that earlier found with equivalent temperature[8].

Adjustment of Effective Temperature for Radiation: Vernon and Warner[9] used the globe thermometer reading in place of the dry-bulb temperature in determining effective temperatures, and they also adjusted the observed wet-bulb temperature so that they effectively used the true vapor pressure.

When during the war the author was asked about such a radiation adjustment he advocated the use of the globe thermometer figure, but in order to reduce complications he recommended that the observed wet-bulb temperature be used. He was convinced that any error so introduced would be slight. The effective temperature thus simply adjusted became known as *corrected effective temperature* and was adopted for use in the Royal Navy[10]. From the later information of Humphreys and his colleagues[11] the author was able to calculate that on the whole the correction seemed to work very well[12], but it was not for a few years that complete validation was given by the work of the Tropical Research Unit[13]. With various elevations

* Throughout the paper s.d. = standard deviation.

of MRT the physiological reactions were closely related to the corrected effective temperature.

The improved correlation with comfort votes when the radiation correction is applied has already been referred to by the author.

COMFORT ZONES

From the results obtained in a study of the thermal comfort of a large number of people one can arrive at a temperature or effective temperature which represents the optimum for those people. In a study by the author of factory workers in Great Britain[7] it was determined that the optimum effective temperature for winter conditions was 61 F, and in an earlier investigation Yaglou and Drinker[8] found an optimum summer figure of 70.5 F. People vary in their thermal demands, however, and the engineer needs some margin of tolerance, so it has become the practice to specify comfort zones rather than exact optimal conditions.

There are wide divergences between the comfort zones that have been determined for various places and seasons. The differences are probably due to a variety of causes. In temperate climates habits of clothing account for considerable differences[14], and habituation or inurement to cool or warm conditions plays its part. In the tropics physiological acclimatization may also be presumed to have some effect on comfort.

For the conditions of the hot arid zones of Australia, Drysdale[15] concludes that while people may prefer an air temperature of 75 F, they become accustomed to a feeling of heat and do not feel really uncomfortable until temperature approaches 84 F.

Ellis[16] studied the comfort of Europeans and Asians living in Singapore. Accepting votes of *comfortable, comfortably warm* and *comfortably cool* as signifying reasonable comfort, the comfort zones in terms of effective temperature were:

EFFECTIVE TEMPERATURES (Fahrenheit)

	MEN	WOMEN
European	73–79	72–78
Asian	71–81	76–80

The upper limits of these zones represented air temperatures of 82 to 85 F, which agree closely with Drysdale's figure of 84 F for Australia. In a personal communication Col. N. N. Chopra, of the Indian Army, informed the author that from results obtained in offices in Delhi during the hot months the comfort zone in terms of corrected effective temperature agreed closely with the zones found by Ellis in the humid climate of Singapore.

The criteria on which the limits of the comfort zone are decided are of basic importance. Many of the zones that have been published in the past are those within which at least 50 percent of the votes have been *comfortable, comfortably cool* or *comfortably warm;* presumably all the votes which have not denoted actual discomfort. That means that at the limits of the comfort zone there is an even chance that discomfort will be experienced. It it suggested by the author that such a range is too wide, for a room can scarcely be called comfortable if nearly half the occupants are uncomfortable. In an earlier work by the author he computed com-

fort zones on a similar basis, and was alarmed to find that in terms of effective temperature the zone covered a range of 14 F deg. Scrutiny of the frequency distributions showed that over the range of conditions in which at least 70 percent of comfort votes were exactly *comfortable*, at least 86 percent ranged from *comfortably cool* to *comfortably warm*, and the author adopted that as the basis of the comfort zone which he recommended. Within those zones not more than 1 person out of 6 would feel any discomfort.

It is notoriously difficult to please everybody all the time, and the literature on comfort contains many amusing examples. Nearly a century ago Morin[17] bemoaned such *prejudices* and said that the only way of overcoming them was by the strict observance of well-made regulations. There will probably be wide agreement with this statement provided the regulations are based on an understanding of human requirements.

Leopold[18] has referred to the close tolerance that may be necessary to satisfy a group of people, even though the individual tolerances may be wide. A few years ago the author had outstanding confirmation of such a need. On account of complaints by members of Parliament he was asked to examine the heating and air-conditioning installation of the rebuilt House of Commons. From a record of complaints it became evident that whatever the tolerances of individual members it was necessary to control the temperature within very narrow limits. Accordingly, the author recommended that during ordinary sittings, when the House was not densely occupied, the temperature should be maintained at 67 to 68 F, that during late sittings the temperature should be 1 or 2 F deg higher, and that when the House was crowded the temperature should be reduced by 2 or 3 F deg. Since the temperature has been maintained within narrower limits complaints have been far less frequent.

Leopold advocated the use of discomfort curves rather than graphs showing the proportions of persons who are comfortable. The author entirely agrees with him. Members of the author's Unit nowadays pay much attention to the probable incidence of discomfort under various conditions.

Skin Temperature and Comfort

It was probably variations between individuals that prompted the physiologist Rubner[19], to remark when discussing clothing that it is a mistake to rely on personal feelings of comfort, for, he said, the feeling of comfort is in many people obtuse, undeveloped and variable. In the early years of this century Rubner's disciples[20] were using measurements of skin temperature as indexes of the effects of the thermal environment on comfort. Later, in the United States, Ward[21], observing herself and 4 others, found a close association between skin temperature and comfort votes, and it seemed that by making a series of observations on a given person in equilibrium conditions one could so *calibrate* him that his subjective feeling of warmth could be assessed with reasonable accuracy. But it did not follow that when observations were made on a large number of people it would be possible to estimate the feeling of warmth of any individual with useful accuracy.

This point was explored in the author's observations of factory workers. Skin temperatures were measured on the forehead, the palm of one hand, and on the inner aspect of one foot, and at the same time comfort votes were obtained and environmental measurements were made. When from about 2500 sets of observations partial correlations of warmth sensations with the different skin temperatures

were calculated the total correlation, R, of sensations with the 3 skin temperatures was 0.44, which was lower than the correlation between sensations and either effective temperature or dry-bulb temperature. The root-mean-square error of estimation of comfort votes from the 3 skin temperatures was 0.77 of a unit on the author's scale, so that a person estimated to be exactly comfortable might be feeling too warm or too cool. The predictions were of no practical value, and individual comfort votes could be foretold with slightly greater accuracy from a knowledge simply of the room temperature.

More recently, in work sponsored by the Society[22], comfort votes were more closely correlated with effective temperature or dry-bulb temperature than with the mean skin temperature.

It is probably true that of the various physiological responses to changes in the thermal environment skin temperature is the best index of subjective feelings of comfort so long as gross discomfort is not experienced. But people vary in their demands. Thus, the optimum forehead temperature for my subjects was 93.7 F, yet in 692 observations in which the forehead temperature lay between 93.0 and 93.9 F, 72.1 percent of the comfort votes were *comfortable*, 15.2 percent *comfortably warm* or *comfortably cool*, 10.8 percent *too warm* or *too cool*, and 1.9 percent *much too warm* or *much too cool*. Such individual variations seem to make skin temperature of dubious value as an index of comfort, even though it gives interesting physiological information.

PLEASANT ENVIRONMENTS

The maintenance of the desired effective temperature does not necessarily ensure that the environment will be a pleasant one. Attention must also be paid to the distribution of heat and to the individual factors that make up the thermal environment. Forty years ago Sir Leonard Hill[23] emphasized the importance of adequate and variable air movement, some variability of temperature, and a relatively dry atmosphere. Hill's teaching was based largely on the study of outdoor conditions, and later Dr. Vernon, Dr. Warner and the author[24,25,8] investigated the effects of these factors on indoor comfort. Temperature, humidity, mean air speed, and variability of air speed were measured and at the same time impressions were noted of freshness or stuffiness on a 9-point scale. Excessive temperature or humidity, inadequate speed and uniformity of speed gave rise to feelings of stuffiness, while within limits cool, dry and moving air felt fresh and pleasant.

Air Movement: At a temperature of 65 F or so if the air speed is less than about 15 fpm the atmosphere is likely to feel stuffy, but complaints of draft may arise with speeds in excess of 40 or 50 fpm. Baetjer[26] found that at such a temperature air movement was not perceptible at less than 40 fpm, and it seems that at ordinary room temperatures perceptible air movement may evoke complaints, but that movement which is below the threshold of perception can exert a beneficial effect. In a hot environment just-perceptible movement (120 fpm at 86 F) may not cause complaint, but sedentary people may well dislike much higher speeds.

In the study by Dr. Vernon, Dr. Warner and the author, the variability of air speed was ascertained by making observations at 2- or 2½-sec intervals over a period of 5 min, and the mean variation of speed was expressed as a percentage of the average speed. In a series of observations when the average air temperature was 71 F, and the average air speed 45 fpm, when temperature and average speed were held constant there remained a highly significant partial correlation between

variability of speed and impressions of freshness ($r = 0.43 \pm 0.08$). The average value of this mean variation was 33 percent with a standard deviation of 10.6 percent. It appeared that an increase of 3 percent in the mean variation of speed had about the same effect on subjective impressions as a decrease of 1 F deg in the air temperature. The changes in air speed were random.

Recently, Dr. Turner[27] studied the effects of mild currents of air impinging on the face. In calm air at 66 F a sudden increase of speed from 12 to 30 fpm was enough to produce a just-perceptible feeling of coolness. From earlier observations it can be calculated that if at that temperature the air speed alternated suddenly and with

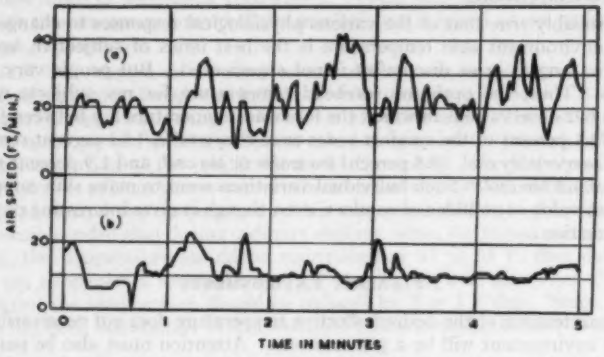

FIG. 2. . . .HOT-WIRE ANEMOMETER RECORDS SHOWING VARIABILITY OF AIR SPEED

random frequency between 12 and 30 fpm, giving an average of 21 fpm and a mean variation of 43 percent, the subjects would have found the air rather fresh, whereas if with the same average speed the mean variation were reduced to 20 percent, say, by alternations of speed between 17 and 25 fpm, they would have had a preference for more stimulating conditions.

Fig. 2 shows the air speeds observed in an office on a June day when the indoor air temperature was 67 F. Curve (a) is a record from the main office ventilated by open windows. The mean speed was 23 fpm with a mean variation of 5.8 fpm. Curve (b) shows the speeds in a small office partitioned off from the rest of the room. The average speed was 12 fpm, with a mean variation of only 2.7 fpm. In the first position the subjects found the conditions neither stuffy nor particularly fresh, but in the second the air felt stuffy and completely stagnant.

Humidity: At room temperatures met with in Great Britain in winter the effect of humidity on the feeling of warmth is small[7], hut there is evidence that changes in humidity which have no appreciable effect on feelings of warmth may yet affect the pleasantness of the environment[28,29].

There was a significant correlation between relative humidity and impressions of freshness when effective temperature was held constant, so that at a given effective temperature one had a feeling of greater freshness at the lower humidities. It is the author's general impression that from the standpoint of comfort a humidity much

above 60 percent is undesirable. Very low humidities also are undesirable, but no discomfort at temperatures of 65 to 70 F was felt when the humidity was about 30 percent.

Warm walls: A century ago some Commissioners appointed by the General Board of Health in England to inquire into the warming and ventilating of dwellings[30] put down as one of the desiderata for comfort that the walls of a room should be at least as warm as the air, and they included cold walls or floors among the conditions that make for discomfort. Although there was no experimental evidence, Dr. Warner and the author were convinced that the Commissioners were right, for on various occasions feelings of stuffiness were experienced when the only possible explanation seemed to be the fact that the MRT was 6 F deg or more below the air temperature.

Later, the author's colleagues[31] studied the reactions of 145 subjects, each of whom was exposed 3 times to a given equivalent temperature (60, 65 or 70 F), once with the boundary surfaces of the room at air temperature, once with the surface temperature 5 F deg below, and once with it 3 F deg above air temperature. The surfaces were maintained substantially at a uniform temperature.

Of those subjects who were exposed to equivalent temperatures of 65 or 70 F, 83 or 84 percent found the warm-wall condition more pleasant than the cold-wall. The warm-wall situation felt distinctly fresher than either of the others, and in the cold-wall condition they felt rather less fresh than when walls and air were at the same temperature. Munro and Chrenko suggested that the greater freshness found with warm walls was due to transient fluctuations in the rate of heat loss from the nasal membranes, for when the walls were warm the air temperature was correspondingly lower and the vapor pressure somewhat lower.

Thermal Gradients: In 1857 it was said[30] that for comfort the floor should be at the highest temperature in the room, while the temperature should gradually decrease toward the ceiling. Little attention seems to have been given to this recommendation; and many heating installations create considerable temperature gradients between floor and ceiling. With such gradients the feet tend to be cool and the head overheated, and even if the head does not feel too warm there is liable to be discomfort which is hard to describe.

In factories the air at head level has often been found to be several degrees, sometimes 10 or 11 F deg, warmer than that near the floor, and with such gradients there have been regularly experienced feelings of stuffiness and the conditions found to be unpleasant. When the gradient over the height of a man has been no more than 2 or 3 F deg one has been very comfortable. Although the author's opinion is based on general observation over many years rather than on critical experiments, he is convinced that the gradient from floor to head level should not exceed 5 F deg, and should preferably be less.

Excessive thermal gradients due to radiant heat are also undesirable. Although an MRT somewhat higher than the air temperature is conducive to comfort when the boundary surfaces are uniformly warm, if the elevated MRT is achieved by local radiation directed toward the head serious discomfort may arise. This is of importance in connection with ceiling heating.

CEILING HEATING

Chrenko[32] has reported the results of intensive experiments on 5 subjects and of further experiments on a group of 150 subjects. The 5 subjects were exposed to the

radiation from a warmed panel (6 ft x 4 ft) or from the whole of a movable ceiling (12 x 18 ft), at different temperatures and different heights, and the larger group to the warmed ceiling at a height of 8½ ft, but with varying temperatures. The air temperature averaged 66 F and the air speed did not exceed 20 fpm.

The subjects reported on their overall state of warmth, and they were also questioned as to their local sensations referred to their heads and their feet. Finally they were asked to say whether the environment was pleasant or not, and to give reasons if it was unpleasant.

The incidence of discomfort was more closely associated with the elevation of MRT at head level due to the overhead radiation (*i.e.*, the actual MRT *minus* the

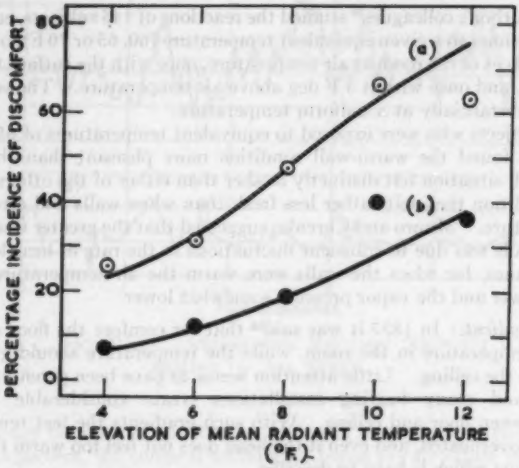

FIG. 3. . . .PERCENTAGE FREQUENCY OF DISCOMFORT IN RELATION TO ELEVATION OF MRT (A) FEELINGS OF UNPLEASANTNESS, (B) UNCOMFORTABLE WARMTH OF THE HEAD

unheated MRT) than to the intensity of the radiation incident on either the forehead or the top of the head.

Fig. 3 shows the results obtained with the group of 150 subjects (441 experiments). The upper curve shows the percentage frequency of complaints of unpleasantness, and the lower one the frequency of uncomfortable warming of the head. Complaints of unpleasantness occurred about twice as frequently as complaints that the subjects' heads were uncomfortably warm.

Subjects who complained of unpleasantness described one or more symptoms to account for their complaints, and an attempt was made to classify these (Table 2).

Nearly 90 percent of the symptoms related to the head or to gradients of sensation. These gradients of sensation which comprised 24 percent of the symptoms

were discussed by Chrenko, who considered that tolerance of a warmed ceiling might well be increased if the floor were also warmed.

Although the average air temperature in these experiments was about 66 F there was some variation between experiments and on some occasions the temperature was as low as 64 F and on others rather more than 70 F. Statistical examination of the results showed that when the elevation of MRT was held constant there was a low but significant partial correlation between air temperature and sensations of warmth of the head. This indicates that at the higher air temperatures tolerance of the radiation from a warmed ceiling declines.

TABLE 2. . . .FREQUENCY OF OCCURRENCE OF VARIOUS SYMPTOMS ASSOCIATED WITH FEELINGS OF UNPLEASANTNESS DUE TO RADIATION FROM OVERHEAD

SYMPTOM	PERCENT OF ALL SYMPTOMS
Warmth on the head	41
Effects on the eyes	5
Other effects on the head	19
Gradients of sensation, *e.g.*, warmth of the head while the feet were chilly	24
Stuffiness	5
General thermal discomfort	6

From his results Chrenko concluded that under the conditions in which ceiling heating is used in Great Britain the elevation of MRT due to the heated ceiling should not exceed 4 F.

FLOOR HEATING

The ancient Greeks, and then the Romans, heated the floors of their buildings by means of the hypocaust. In modern times a notable example of floor heating is that of Liverpool Cathedral. Just after World War I schools in Derbyshire were heated by means of warmed floors, and many observations were made in those schools[33]. Commonly the surface temperature of the floor did not exceed 75 F, and the observers were very comfortable, but when occasionally the surface temperature reached or exceeded 80 F they found the floor very tiring for the feet.

Since that time floor heating has been used in many commercial buildings, and also in dwellings. Various opinions were expressed as to the maximum desirable temperature of a warmed floor, but there was little or no experimental evidence. At a conference in Paris[34] the author expressed the view that the maximum desirable temperature was 75 F or perhaps 77 F, Liese (Germany) agreed with 77 F, while Missenard (France) and Squassi (Italy) allowed 82.5 F. From experiments, Herrington and Lorenzi concluded that floor temperatures above 75 F were undesirable and that the local effect was clearly demonstrable at 80 F.

Chrenko[35] has reported laboratory experiments made on men and women subjects wearing their normal footwear who in different experiments (1) sat with their feet on a heated floor, or (2) walked about on it. The air temperature in individual experiments ranged between 58 and 76 F, averaging 68 F. The subjects were

questioned as to their state of general bodily warmth and they also reported their sensations of warmth of the feet.

Skin temperatures were measured on the ankle, on the sole of the foot at the base of the big toe, and on the dorsal surface of the big toe. The temperature most closely associated with local sensations was that measured on the sole.

For all 4 sets of observations, women sitting or walking and men sitting or walking, the relations between skin temperature of the sole and local sensations were so

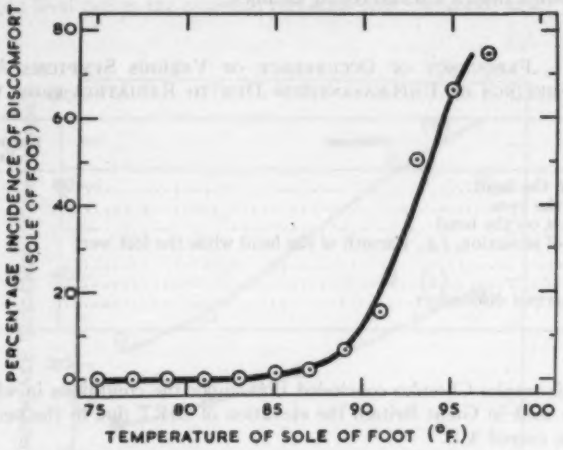

FIG. 4. . . . PERCENTAGE INCIDENCE OF UNCOMFORTABLE WARMING OF THE FEET AFTER 60 MIN EXPOSURE TO HEATED FLOOR

similar that it was justifiable to group the results. For the grouped results (801 experiments) the correlation coefficient was + 0.72 ($P < 0.001$).

Fig. 4 shows the relation between the incidence of discomfort and the skin temperature of the sole of the foot. Some subjects felt discomfort when the skin temperature exceeded 86 F, and when it reached 93½ F, 50 percent were uncomfortable.

Local skin temperature is influenced by the general thermal state of the body as well as by local exposures, so that the temperature of the sole of the foot is affected by the air temperature as well as the temperature of the floor. Regression equations in which t_s is the skin temperature of the sole, t_a the air temperature, and t_f the floor temperature (all in F deg) for the 4 groups were:

	SITTING	WALKING
Women	$t_s = 0.494t_a + 0.351t_f + 25.7$	$t_s = 0.451t_a + 0.488t_f + 18.1$
Men	$t_s = 0.603t_a + 0.379t_f + 14.8$	$t_s = 0.280t_a + 0.813t_f + 0.5$

The regression constants are subject to some error and the actual values should not be stressed to greatly. It is significant, however, that when the subjects walked and thus made firmer contact with the floor the effect of floor temperature was greater than when they sat.

The highest floor temperature at which no discomfort was experienced was 77 F. At the room temperatures commonly maintained in Great Britain considerable warming can be achieved with such a floor temperature, which, according to these results, is unlikely to occasion even mild discomfort.

The skin temperature on the sole of the foot was affected by the air temperature as well as the floor temperature. In the sitting subjects, both men and women, a change of 1 F deg in the air temperature had a greater effect than a similar change in the temperature of the floor, but when the subjects were walking the floor temperature exerted the greater effect. This effect of air temperature shows that a floor that would be uncomfortably warm in a room heated to one temperature might be quite acceptable in a cooler room.

Nevins and Flinner[30] have reported experiments in which they found no significant correlation between comfort votes and the temperature of the floor. The comfort votes presumably related to general bodily warmth, and in that event the lack of correlation is not surprising. Mr. Chrenko informed the author that out of 666 experiments in which, apart from local sensations referred to the soles of the feet, the subjects were otherwise quite *comfortable*, in 21.5 percent instances the feet were uncomfortably warm, and in a further 56 percent the feet were said to be *comfortably warm, i.e.*, while these subjects did not suffer actual discomfort they would have preferred their feet to be somewhat cooler. This local discomfort when conditions are otherwise satisfactory is similar to what the author encountered many years ago in connection with gradients of air temperature[7], when factory workers complained that their feet were cold but insisted that they would not like the general temperature of the workroom to be any higher.

ACKNOWLEDGMENT

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DISCUSSION

L. P. HERRINGTON*, New Haven, Conn.: For a long time, I have followed with the closest interest Dr. Bedford's informed analysis of the human factor in thermal engineering. Several questions which he has emphasized concerning the element of comfort measurement have been the subject of experimental concern for me, and this Session offers an opportunity to discuss the experience in these matters of the Pierce Foundation's Laboratory of Hygiene. We are certainly not in any fundamental disagreement on these problems. However, I do belong to the school of thought which feels that the only items heating and air-conditioning engineers can deliver to an individual, or a group, are an average air temperature and velocity, a mean radiant temperature, and a controlled humidity. Likewise, the only items the human factor can deliver in this situation are a mean body temperature and a skin temperature topography.

In our common experience with groups we frequently find ourselves confronted with an objectively determined thermal environment controlled within a mean comfort zone, an objectively determined body temperature and skin temperature topography, and the disturbing fact that a large number of people report that they are uncomfortable in this region of mean thermal comfort for the group.

Some of the experimental work quoted by Dr. Bedford points up this moderate correlation (in studies of working populations) between objective factors such as partial skin temperatures, objective environmental thermal measures and comfort votes. This difficulty is complicated by the fact that one could quote other reliable research (Nevins and Flinner[36]) to show that the correlation between comfort votes and such factors as air temperature, effective temperature, operative temperature, and air temperature plus MRT, is quite high (of the order of 0.90 as contrasted with 0.40 to 0.50 reported in one of the studies discussed in Dr. Bedford's paper).

Actually, the data of both sets of experimenters appeal to your discusser as being quite reliable. Nevins and Flinner standardized age and activity, and worked in a range from 65 to 85 F over which the relative thermal sensation element is predominant for even the unpracticed subject's psycho-physical judgment of thermal comfort. However, when one moves from these conditions to a practical industrial group of 2500 workers with activity, age, clothing, and spot environmental conditions of less standardized character, the relation of objective measures of thermal effect or ambient conditions to reported thermal comfort is greatly reduced. The principal point here is that one should not allow this reduction of correlation to cast doubt on the basic human engineering value of objective measures. In a real sense the thermal comfort report problem in a narrow range is comparable to the tailor's problem. The mean suit of clothing is exactly right,

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* Director, John B. Pierce Foundation Laboratory of Hygiene.

apart from a tuck or two, for very few people. In addition, there are cultural and experience differences in the preference of different individuals for a condition either of slight body cooling or slight body heating in relation to a comfort reaction.

Some of my medical associates have humorously pointed out that narrow range comfort votes may have deep emotional bias. The entry into this world involves the novel experience of a 25 F cold shock, under usual conditions. Those born under *proper* circumstances then get a bath. It has been suggested quite seriously that part of the thermal engineer's comfort problem is due to the fact that some of the lazy ones liked the cold shock, and that some of the tense ones liked the bath.

In dealing with these problems, which implicate what could be called a thermal bill of rights (the right to vote in an unpracticed manner on a sensation of transient character), I prefer an experimental preface which attempts to quantify the exact effect of differences in age, sex, clothing, and activity on heat production, and the resultant total skin temperature topography; in relation to the integrated thermal forces of the environment (ASME Heat Transfer Papers No. 57-SA-5, June 10, 1957; No. 58-A-181, December 5, 1958) taken in conjunction with concepts of clothing thermal resistance (*Yale Journal of Biology & Medicine* Vol. 19, No. 4, 1947) and shifts in the zone of individual thermal neutrality related quantitatively to activity differences.

The next step is to relate the resultant skin temperatures corrected for the foregoing processes to the thermal judgments of highly practiced subjects (*American Journal of Physiology*, Table 2, p. 34, Vol. 124, No. 1). If it is then found that objective factors disperse thermal neutrality for the individuals of a uniform group over a range of 8 to 10 F deg under highly controlled conditions of calorimeter measurement and practiced conditions of sensation judgment, one should not be too much concerned to find that less practiced thermal comfort judgments in less uniform practical environments are distributed over a 10 to 15 F deg range for more heterogeneous groups.

From the equations of the ASME reference just mentioned it can be shown that spreads of ±15 percent in basal heat production can be responsible for a difference of at least 2.9 F deg of skin temperature in individuals of the same age and sex under standard environmental conditions of approximate comfort. Reliable measurements show that if the group is mixed, a difference of about 10 percent in heat production is sex correlated. At a thermal head of approximately 400 Btu per hour we ascribe to 6 to 7 pounds of male clothing a thermal protection value of 15 to 16 F deg. Male clothing actually varies from approximately 6 to 8 pounds, and is generally 50 to 60 percent of this value for the female. It can likewise be shown from standard tables of metabolism that a heat production difference of ±47 percent exists between a male of 18 years, and a female of 62.5 years in age. Individuals combining the extremes of these characteristics may be in the same group. Without considering the reasonable possibility of differences of 10 to 30 percent in socalled quiet activity heat production, it can be shown that the combination of genetically low heat production at standard age and maximum sex-qualified age difference (18 to 62.5 years) in heat production can account for an equational prediction difference (ASME references) in mean skin temperature of at least 5.8 F deg. From the data of the *American Journal of Physiology* reference, it can be shown that in 3-hour calorimeter exposures of uniformly clothed male subjects of standard age a mean thermal comfort vote of unpleasantly warm at 75.9 F shifts to a mean vote of unpleasantly cool at 62.1 F. Correlated with this shift in practiced thermal comfort vote is a change of 5.04 F deg in mean skin temperature.

A gross conclusion from these data suggests that under standard environmental thermal conditions,—age, sex, dress, and genetic constitution, may produce a thermal sensation or comfort effect in terms of mean skin temperature, comparable to the effect in a single standard subject of a shift of 14 F deg in environmental temperature. To this biological range in the comfort zone for the heterogeneous groups of commerce, industry, and public assembly, must be added the effects of clothing differences and the inevitable activity differences in any practical heating or air-conditioning circumstance.

Dr. Bedford has made many of these effects clear, and I will not labor the point, except to say that while re-study of the useful comfort scales may quantitatively improve

the corrections for radiation or humidity (in certain ranges) one should not expect the ASHAE Laboratory to produce a scale which will compress the variable human heat production characteristic and social diversities of clothing and activity into a very much narrower range. Personally, I feel that it is a minor marvel that the current comfort scales work as well as they do.

The fine discrimination of such scales for average application is, in the end, determined by statistical constants which apply to physiological and dress characteristics, provided activity is standardized. If one refers to reliable sources for information on the tailor's problem (variation in anatomical dimensions) he may realize that the zone of universal thermal comfort is as broad as the range of tailoring variations required to fit a major portion of the population.

I do not feel that these facts discussed so informatively by Dr. Bedford introduce any impossible criticism of the comfort scales. It seems to me that we should correct our natural reaction to their inconvenient spread with a glance at the practical commercial pressures which make the circumstance important to the engineer.

As you have no doubt gathered in this discussion, I am willing to define thermal comfort for the individual in terms of a stabilized body temperature of normal value and a skin temperature topography. For men of normal metabolism, wearing 6 to 7 pounds of clothing, seated quietly for 3 hr, at age *circa* 40 (the approximate average of the American population), this stabilized skin temperature is close to 92 F as a grand average of 15 body points, the forehead temperature being close to 93 to 94 F, with the skin temperature reaching an extreme low value of 75 to 80 F on the extreme portions of the foot. The politics of the body physiologic do not always permit an individual so stabilized in a temperature sense to vote himself comfortable, but the evidence is very good that a thermal engineer could not improve his strictly thermal status. A portion of my considerable concern with this objective emphasis is due to the fact that at the moment both civil and military problems are requiring the determination of thermal conditions of minimal temperature stress for quite unusual circumstances. In these problems one is forced to give a larger weight to objective calorimeter data than to the social factors which influence to a degree the application of thermal comfort criteria to ordinary but indeed commercially important civil problems.

In the same vein, I would comment that attention might well be shifted somewhat from the search for the universal temperature zone of maximum comfort and minimal spread. It is my persuasion that current problems recommend concentration of more effort in the direction of improving the objective human engineering data required for the custom design of thermally neutral environments for groups with known age, sex, clothing, and activity specifications. Further, I think that this design information should direct its quantification not only to the coefficients of ambient factors required for optimum thermal comfort, but should likewise generate the technical data appropriate for the design of specified degrees of thermal stress. This I believe to be the challenging direction of advance for ASHAE from the standpoint of engineered thermal environments of broad spectrum design adequacy. Certainly no investigator has done more than Dr. Bedford, and his associates, in clarifying the multitude of problems that confront us in this expanding field of engineering science.

C. S. LEOPOLD, Philadelphia, Penn.: Dr. Bedford, in one of your writings you have stated that "the temperature should be maintained as low as is consistent with comfort." I wish you would elaborate on that, if you will.

AUTHOR'S CLOSURE: I am delighted to see Dr. Herrington here, for it is more than 10 years since we last met. On the whole, I think I agree with much that he has stated. Age, sex, clothing, and activity all exert their influence on thermal demands, and they account very largely for considerable divergences between what appear to be the environmental temperatures most preferred by different people. I also agree that lack of practice in expressing thermal judgments is likely to be another source of variability in the results obtained in studies of comfort.

Dr. Herrington has mentioned that the room temperature for thermal neutrality—which may perhaps be regarded as the optimum temperature—may vary over 8 to 10 F deg in a uniform group of subjects. Fortunately, it is not necessary to provide for each person a fixed optimum temperature. For each person there is a range of a few degrees within which he is comfortable. The engineer seeks to provide conditions which will best meet the conflicting demands of a variety of people, and presumably that is the original reason for assessing comfort zones. There must be compromise so as to give the highest possible degree of satisfaction to the greatest number of people.

I have much sympathy with Dr. Herrington's final remarks. Undoubtedly aged people need a warmer environment than satisfies younger and more vigorous people, and a few years ago Dr. E. F. DuBois, in a paper presented to the Society, suggested the provision of local warming for grandma. Shortly afterwards I mentioned Dr. DuBois' suggestion at a meeting in England, and Prof. C. G. Douglas, an eminent physiologist, remarked: "My grandma just put a shawl around her shoulders."

Mr. Leopold has referred to a statement of mine that "the temperature should be maintained as low as is consistent with comfort." That is a qualification of a statement made 40 years ago by Sir Leonard Hill. He said that a room should be cool rather than hot. Many years ago, as a young disciple of Sir Leonard, I used to work in a cool room. Now I like rather more warmth, but I still try to avoid being overwarm. I am convinced that so long as one is not uncomfortable a cool room is more conducive to alertness than is a warm one. I am satisfied that my output of work is likely to be a little higher with a temperature of 65 or 66 F than with one of 70 F.

1654

No. 1654

METABOLIC PATTERNS IN STRESSFUL ENVIRONMENTS†

By NELLO PACE*, BERKELEY, CALIF.

AMONG all of the life forms on earth, man has exhibited by far and away the greatest capacity for the successful penetration of harsh environments of all descriptions. Thus he has scaled the highest peaks of the land masses, he has gone for weeks many feet below the surface of the seas, he lives for months in the South Polar region during the Antarctic winter when temperatures range from −70 F to −120 F and colder, he has entered bake ovens at 260 F and remained for minutes at a time, and he has sailed aloft for hours into the outer reaches of the atmosphere a dozen or more miles above Mt. Everest, itself rising 5½ miles above sea level. Today he stands poised for a leap beyond the confines of earth's gravity into the greatest venture of all time . . . space flight.

It is clear that man's outstanding achievements as an otherwise rather ordinary terrestrial mammal depend in large measure upon his unique ability to reason and to transmute his imaginings into practical reality. But while intellect has borne the ever growing fruit of technology, patterned in the form of the engineering sciences, man seems perennially to be too impatient to await its full ripening before thrusting ever farther beyond his natural ecological niche.

There has developed the field of environmental physiology, therefore, in which it is sought to evaluate the limits of tolerance for, or the capacity to adapt to, the multitudinous factors which can comprise man's ambient. Merging with this delineation of extremes of endurance is another objective of the environmental physiologist . . . the definition of the optimal conditions for human existence, or the determination of *comfort zones*. The physiologist, however, is not content with the mere cataloging of performance specifications for the human organism, useful

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though this may be, but seeks in addition to understand the interplay of the underlying mechanisms and processes within the body that determine adaptability to changes in environment.

It is to this last aspect of environmental physiology that the author wishes to address himself on the present occasion, because it is here that physiology can make its greatest contribution ultimately by serving an integrative function in biology generally. Also, of more immediate interest to this audience, some inkling may be gained at present of how the human being responds when he enters stressful environments.

Much of what the author has to say today traces directly from the greatest generalists of earlier generations of physiologists such as Claude Bernard, Walter B. Cannon and Sir Joseph Barcroft, as well as from such eminent contemporaries as Hans Selye of the University of Montreal, Gregory Pincus and Hudson Hoagland of the Worcester Foundation, Roger J. Williams of the University of Texas, and George W. Thorn of the Peter Bent Brigham Hospital.

I have also taken advantage of the occasion to interject some of the thinking and some of the data that have been developing in the Environmental Physiology Laboratory in Berkeley during the past few years. What is shown here is the result of the highly laudable efforts of a number of workers in our group, too many to enumerate, but the author does wish to mention in particular Prof. Paola S. Timiras of the Department of Physiology, as an active and capable collaborator in these studies.

CONSTANCY OF INTERNAL ENVIRONMENT OF THE BODY

One of the fundamental concepts in physiology, developed by Claude Bernard nearly a century ago[1], is that of the constancy of the internal environment of the body. According to his view, a great many if not all of the body processes function to preserve or restore the cellular environment in the face of external environmental changes so that the optimal conditions for life may be maintained. Thus there is the complex counterbalance between insulin, adrenaline, the adrenocortical glycotropic hormones and the body stores of glycogen, which serves to maintain the blood sugar at a relatively uniform level. Similarly there is the pattern of changes involving the amount of blood flow through the skin, the initiation of sweating or of shivering, and the secretion of thyroid hormone . . . all aimed at the maintenance of deep body temperature within a narrow range. Such examples can be multiplied many fold.

The principle of the maintenance of uniformity of the characteristics of the body fluids through adaptive mechanisms was termed *homeostasis* by Walter B. Cannon, who did much in the early part of this century to expand the work of Claude Bernard. Cannon was also one of the first to introduce the concept that our bodies constitute open systems engaged in continuous exchanges with the external environment[2]. In other words, homeostasis represents a remarkably well controlled complex of steady state levels that is maintained in the face of wide fluctuations in input and outgo of matter and energy between organism and environment.

Attention may next be directed to the quantitative aspects of homeostasis, and it is readily seen that the greater the deviation, chemically or physically, of the ex-

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[1] Exponent numerals refer to References.

ternal environment from the optimum, the greater is the adjustment or adaptation which the organism must make in order to preserve homeostasis. Thus one arrives at the concept of *physiological stress*, a term much used and abused, yet a convenient one. In the present context it can be defined as representing the flux density or *flow through* involved in the maintenance of homeostasis. It is a measure of the degree of insult or severity of the stressor to which an organism is subjected, and hence of the degree of adjustment that must be made. On such a basis, then, it may be said that the organism can tolerate up to a maximum degree of stress before failure will occur, and that the degree of stress tolerable will depend upon the kind of stressor

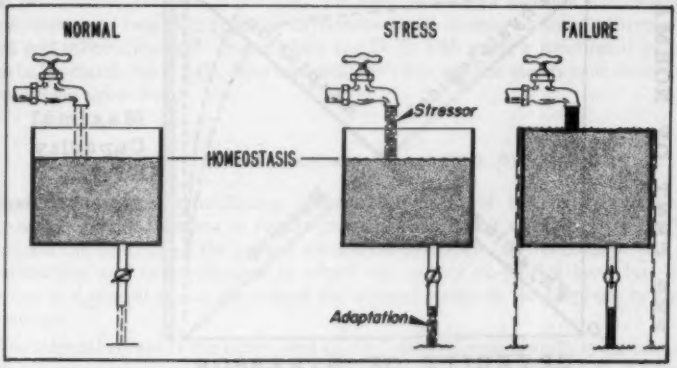

FIG. 1. . . .STEADY-STATE ANALOGY TO ILLUSTRATE THE DIFFERENCE BETWEEN PHYSIOLOGICAL STRESS AND PHYSIOLOGICAL FAILURE. HOMEOSTASIS IS ILLUSTRATED AS THE CONSTANT FLUID LEVEL IN THE FACE OF INCREASED FLOW THROUGH THE SYSTEM. FAILURE OCCURS WHEN THE CAPACITY OF THE BODY ADAPTIVE MECHANISMS IS EXCEEDED

involved and the adaptive mechanisms available to the organism to cope with the particular stressor.

Some of these ideas are summarized diagrammatically in Fig. 1. Like all analogies, it does not bear too literal an interpretation, but does serve to illustrate the dynamic relationships involved. Needless to say, the actual situation is far more complicated.

TASK OF THE PHYSIOLOGIST

The task of the physiologist in analyzing the relationship between organism and environment can be outlined against the concepts presented in the foregoing. Basically this consists in determining the physiological cost to the organism of adapting to a new environment, measured in terms of maximal adaptation capacity. *A priori* it might appear, if the body mechanisms are diverted to the task of adjust-

ing to environmental change, that the performance of the organism as a whole is likely to be less effective.

Much work has been done by many investigators in attempts to measure deterioration of performance, particularly that involving the central nervous system, under many environmental conditions of varying degrees of severity. The hope that performance in tests of reaction time, memory, arithmetic computation and the like might serve to make comparative evaluations of environmental condi-

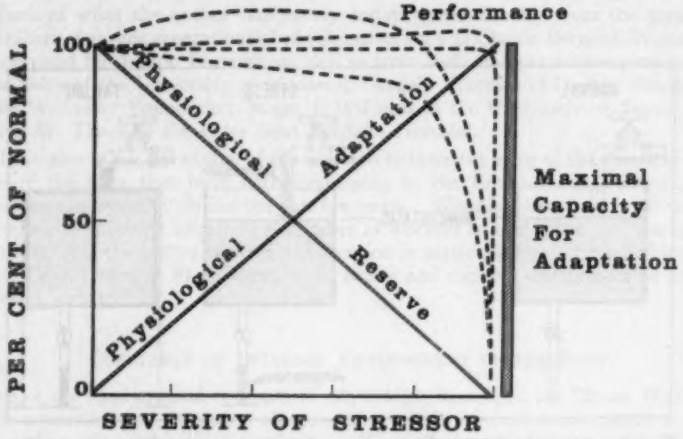

FIG. 2. . . .SCHEMATIC REPRESENTATION OF THE TYPES OF RELATIONSHIPS THAT FREQUENTLY OCCUR BETWEEN CENTRAL NERVOUS SYSTEM PERFORMANCE TESTS, DEGREE OF PHYSIOLOGICAL ADAPTATION INVOLVED, DECREMENT IN PHYSIOLOGICAL RESERVE, AND SEVERITY OF ENVIRONMENTAL STRESSOR

tions has largely waned, however, and much has been made of the role of such difficultly definable psychological concepts as motivation.

Sir Joseph Barcroft 25 years ago[3] clearly perceived in Claude Bernard's famous dictum *the fixity of the internal environment is the condition of the free life* that protection of the central nervous system is the prime function of homeostatic mechanisms, so that as long as homeostasis is maintained, brain function is not impaired. Conversely, when homeostasis is lost, brain function is the first casualty. Thus under the present set of definitions it is easy to see how it is that higher integrative performance may well be maintained in the face of even severe physiological stress.

Fig. 2 is a schematic representation of the general relationships which have frequently been found between performance and physiological adaptation as a function of increasing severity of a given stressor. Some of the features to be noted are the actual improvement sometimes seen in performance under stress . . . the *facilitation* of the psychologist . . . and the fact that as maximal adaptive capacity is

reached, performance deteriorates precipitously. In order to gage quantitatively the severity of a stressful environment, then, it would appear more meaningful to measure the degree of adaptive response evoked than to examine performance involving higher brain function.

On the other hand, the adaptive response to a stressor, as a general rule, is exceedingly complex and involves many of the body organ systems. Further, different kinds of stressors evoke different types of responses. Finally, the duration of exposure frequently determines the nature of the response pattern, so that acute effects must be differentiated from chronic effects. For example, sudden exposure to elevated ambient air temperatures leads to quite a different set of physiological responses than does exposure to acutely reduced oxygen partial pressure in the air, and heat acclimatization is uniquely different from high altitude acclimatization. How, then, can two such types of environments be compared among themselves or with still other stressors? Some years ago D. B. Dill made a most lucid juxtaposition in his classic book *Life, Heat and Altitude*[4], but left the question of direct, literal comparison unresolved.

CONCEPT OF THE GENERAL ADAPTATION SYNDROME

Perhaps the most stimulating, and certainly one of the most controversial, approaches to the problem in recent years has been that of Hans Selye[5], who introduced the concept of the *general adaptation syndrome*. This concept has many ramifications and extrapolations to which one cannot do justice here, but it does involve as a central theme the role of the adrenal cortex in the response to stressful situations.

The adrenal cortex is the outer shell of the 2 tiny adrenal glands, lying just above each kidney in man, and is the site of production of a variety of important hormones that regulate a number of metabolic processes. These include the glycotropic substances cortisone and hydrocortisone, the electrolyte regulating hormone aldosterone, and a variety of androgenic substances involved in the development and maintenance of male secondary sex characteristics and body protein synthesis. The precise details of how the adrenocortical hormones function are still not known, but their fundamental importance to the life process is unquestioned.

Two other points should be made in this endocrinological thumbnail sketch. The production of the glycotropic hormones cortisone and hydrocortisone, important in carbohydrate metabolism, appears to be under the control of yet another hormone, adrenocorticotropic hormone or ACTH, which is one of a number produced by the pituitary gland situated at the base of the brain. This relationship has been delineated by the coining of the phrase *pituitary-adrenal axis* because of its ubiquitous role in stress physiology.

The last point concerns the central or medullary portion of the adrenal glands. The adrenal medulla is the site of formation of the hormone adrenaline or epinephrine, and is regulated by the sympathetic nervous system. The sympathetic nervous system, adrenal medulla complex, or sympathico-adrenal system of Cannon[2] has long been recognized as representing one of the body's first lines of defense against environmental disturbances, and the *alarm reaction* is a classic immediate response pattern.

It is the thesis of Selye that whatever the form of stressor applied to the organism may be, at least one component of the response will comprise the pituitary-

adrenal axis and the sympathico-adrenal system, and that if the condition of stress persists long enough, the organism will begin to suffer from over-production of hormones, so that *diseases of stress* will result. While much of Selye's argument is the subject of debate, it can at least be said that he has been responsible for directing widespread attention to the role of the adrenal cortex in stress. At the same time, however, it must be pointed out that a number of equally outstanding, it less prolix, contemporary investigators have also been responsible for the recent rapid advances in this area.

PROBLEM OF COMBAT FATIGUE

Such was the situation in the summer of 1952 when the author found himself on active duty with the Navy and his services as an environmental physiologist were requested to lead a joint Army-Navy team in a study of combat in Korea as an environment for infantrymen. As is frequently the case in the military, the project was attended by a considerable measure of urgency, and less than a month was available for preparation. The principal target of the study, to be carried out in the front lines, which at that time had been relatively well stabilized, was the poorly delineated problem of combat fatigue. The group to make this study was designated the *Fatigue and Stress Team*, otherwise known as *FAST*, and was composed of 8 physiologists and biochemists, together with 4 psychologists and one psychiatrist. In spite of many allusions to *fast company* and in spite of numbering 13, the group was able during the fall of 1952 to derive a comprehensive set of objective physiological data on the effects of severe combat on human beings for the first time, and without losing an investigator.

To make a long story short, while preparing for the study it occurred to me that inasmuch as the classical physiological techniques previously used in examining environmental effects, such as the recording of skin and body temperature, blood pressure and work capacity, might prove awkward to apply and difficult to interpret under combat conditions, it might be fruitful to look for evidence of metabolic responses to the stressor. Attention naturally turned to the adrenal cortex because of the important advances that had been and were being made at that time. The isolation and chemical identification of some of the adrenocortical hormones had been accomplished, and reliable chemical methods for their estimation were being developed. In addition, *FAST* had among its members 2 investigators from the Worcester Foundation in Massachusetts who were prepared to measure blood eosinophil and urinary 17-ketosteroid levels, together with several physiologists from the Berkeley laboratory who were competent biochemists as well.

The reasoning applied to the problem was that by chemical examination of blood and urine specimens it should be possible to obtain evidence of metabolic adjustments that individuals might make to the environmental stressor, and that some estimate of the severity of the stressor might be made from the degree of the metabolic response evoked. Thus, by comparison of metabolic patterns or *profiles* obtained under control and stressful conditions, an evaluation of the physiological effect could be made. Such metabolic response patterns could also be compared with the tests of cerebral performance to be administered by the *FAST* psychologists.

It was decided, therefore, to measure constituents of blood and urine that related to adrenal cortical function. Some of the constituents were measured in a field

laboratory established in Korea and the remainder of the analyses were carried out upon return to the United States. Fig. 3 shows the relationship of the metabolic parameters examined to the physiological functions with which they are associated. If nothing else, the figure shows the complexity of just a few of the inter-locking homeostatic mechanisms that must be unraveled. However, it also shows the need for making simultaneous measurement of a number of parameters if overall function is to be assessed.

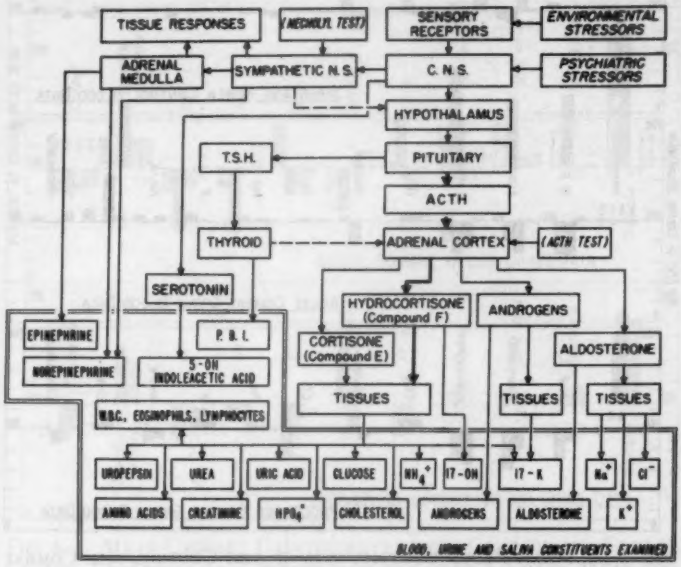

FIG. 3. . . .GENERALIZED PHYSIOLOGICAL RELATIONSHIP BETWEEN THE STRESSOR AND THE METABOLIC EFFECTS THAT MAY BE MEASURED BY CHEMICAL ANALYSIS OF VARIOUS BODY FLUIDS. SOME OF THE CONSTITUENTS THAT CAN BE DETERMINED ARE SHOWN ENCLOSED BY THE DOUBLE LINED BOX AT THE BOTTOM OF THE FIGURE

Many investigators in this field have measured one or more of the items listed here in a variety of circumstances, but the concept of a metabolic pattern approach was relatively new. Roger Williams of Texas had made a somewhat similar approach[6], but with emphasis on amino acid metabolism rather than adrenocortical function. More recently his attention has turned to an examination of the underlying significance of individual differences in metabolic characteristics within a given population[7], whereas our interest has been focused on the changes occurring within a given group as a result of exposure to a particular environmental stressor.

One other novel aspect of the present approach has been the emphasis upon examining the changes in excretion rate of substances in the urine, rather than changes in the circulating blood level. It has been the premise that homeostasis

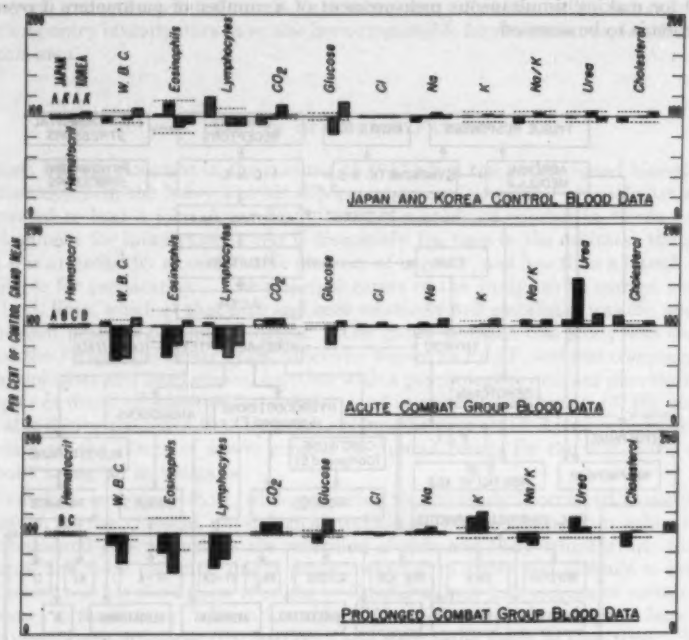

FIG. 4. . . .MEAN BLOOD CONSTITUENTS OF THE CONTROL AND COMBAT GROUPS, EXPRESSED AS A PERCENTAGE OF CONTROL GRAND MEAN FOR EACH CONSTITUENT. UPPER SECTION REPRESENTS TEST DATA, A, AND RETEST DATA, A′, OBTAINED SEVERAL DAYS LATER, FOR JAPAN AND KOREA CONTROL GROUPS. CENTER SECTION REPRESENTS DATA FROM ACUTE COMBAT GROUP OBTAINED: 12 HR BEFORE COMBAT, A; 12 HR AFTER COMBAT, B; 5 DAYS AFTER COMBAT, C; 22 DAYS AFTER COMBAT, D. LOWER SECTION REPRESENTS DATA FROM PROLONGED COMBAT GROUP OBTAINED: 12 HR AFTER COMBAT, B; 10 DAYS AFTER COMBAT, C. DASHED LINES REPRESENT STANDARD ERROR OF CONTROL GRAND MEAN FOR EACH BLOOD CONSTITUENT MEASURED

tends to be operative in stress, as outlined earlier in this paper, so that blood levels may remain relatively constant, whereas metabolic adjustments that are likely to occur are probably reflected more obviously in changed excretion rates in the urine. Measurement of excretion rates by way of the urine also has the great advantage

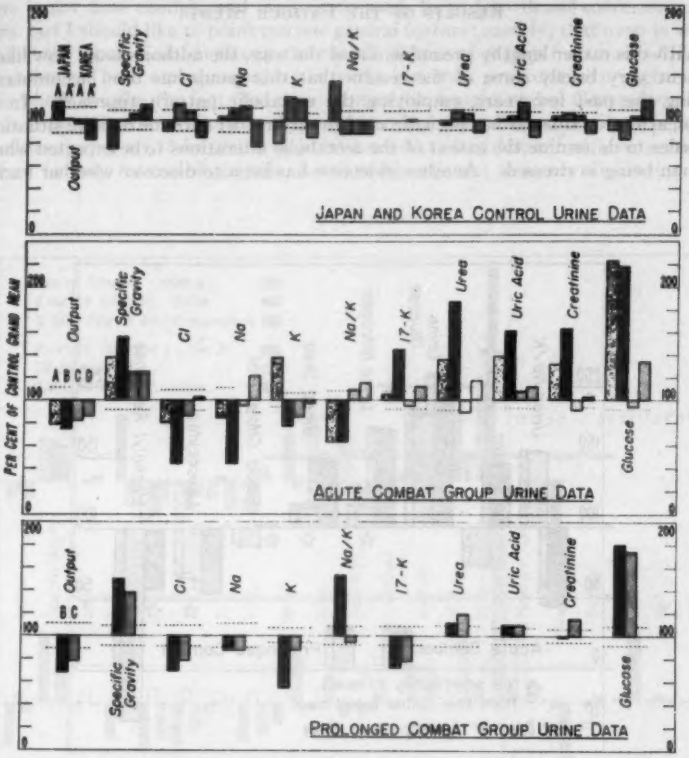

FIG. 5. . . .MEAN URINARY CONSTITUENTS OF THE CONTROL AND COMBAT GROUPS, EXPRESSED AS A PERCENTAGE OF CONTROL GRAND MEAN FOR EACH CONSTITUENT. UPPER SECTION REPRESENTS TEST DATA, A, AND RETEST DATA, A′, OBTAINED SEVERAL DAYS LATER, FOR JAPAN AND KOREA CONTROL GROUPS. CENTER SECTION REPRESENTS DATA FROM ACUTE COMBAT GROUP OBTAINED: 12 HR BEFORE COMBAT, A; 12 HR AFTER COMBAT, B; 5 DAYS AFTER COMBAT, C; 22 DAYS AFTER COMBAT, D. LOWER SECTION REPRESENTS DATA FROM PROLONGED COMBAT GROUP OBTAINED: 12 HR AFTER COMBAT, B; 10 DAYS AFTER COMBAT, C. DASHED LINES REPRESENT STANDARD ERROR OF CONTROL GRAND MEAN FOR EACH URINARY CONSTITUENT MEASURED

that metabolic alterations within the organism may be integrated during the period of the urine collection, that is for minutes or hours at a time, whereas a blood sample indicates the level of a constituent only at the time of sampling. Both blood and urine samples have been taken, however, as the opportunity has permitted.

RESULTS OF THE FATIGUE STUDY

With this rather lengthy preamble out of the way, the author should now like to present very briefly some of the results that this group has been accumulating during the past few years, employing the metabolic pattern approach. In the main, attention thus far has been directed toward rather extreme real life situations, in order to determine the extent of the metabolic alterations to be expected when a human being is stressed. Another objective has been to discover whether various

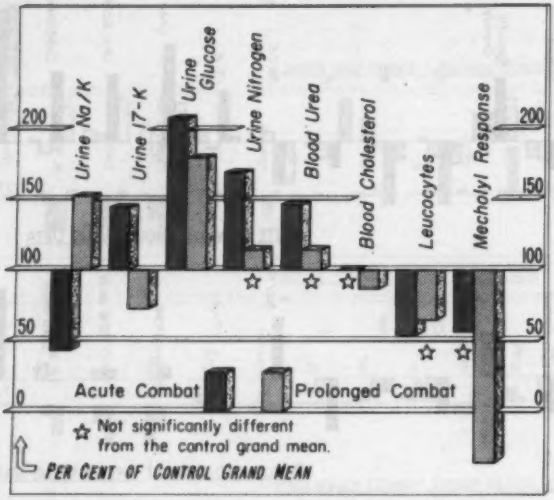

FIG. 6. . . .SUMMARY OF MEASURES WHICH STATISTICALLY WERE SIGNIFICANTLY DIFFERENT FROM CONTROL VALUES SOON AFTER COMBAT IN EITHER OR BOTH ACUTE AND PROLONGED COMBAT GROUPS. VALUES ARE EXPRESSED AS A PERCENTAGE OF CONTROL GRAND MEAN FOR EACH MEASURE

kinds of environmental stressors produce metabolic alterations that differ in quality as well as in quantity, or whether a uniform pattern of metabolic response is seen for all stressors.

Fig. 4 shows the kind of results obtained from blood analyses during the Korea combat study[8]. The data shown represent mean values of various blood constituents for 4 groups of soldiers . . . 2 control groups and 2 groups exposed to intensive combat. The mean value for each constituent is expressed as a percentage of the normal mean value as obtained from a standard reference source such as Albritton[9].

There is not time enough, and there is no need, to go into detail concerning this figure, but I should like to point out one general feature; namely, that even in what is called the Acute Combat Group the overall pattern of change from normal is not impressive. There are changes, to be sure, but they are relatively slight. In other words, despite the fact that these men in the Acute Combat Group were survivors from an attack infantry company that had just suffered some 61 percent casualties during an 18-hr period of intensive action, and hence had been severely stressed by any reasonable criteria, homeostasis was not seriously violated.

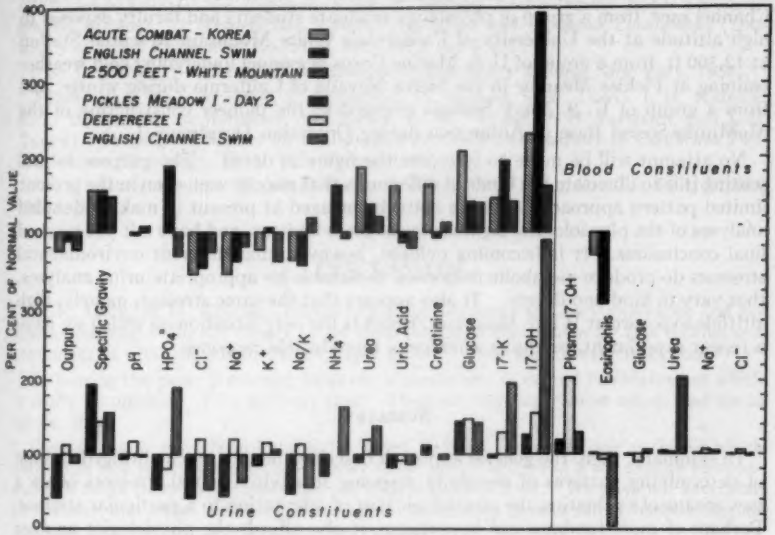

FIG. 7. . . .SUMMARY SHOWING MEAN BLOOD LEVELS AND URINARY EXCRETION RATES FOR VARIOUS CONSTITUENTS FROM GROUPS OF MEN EXPOSED TO A VARIETY OF ENVIRONMENTAL SITUATIONS, EXPRESSED AS A PERCENTAGE OF THE NORMAL VALUE FOR EACH CONSTITUENT

In comparison, Fig. 5 shows the results of the urine analyses from these same groups. Again, the data represent mean values of the excretion rate of each constituent, expressed as a percentage of the population mean. In contrast to Fig. 4, there is clear indication of marked change in the excretion pattern as a result of the combat experience.

Fig. 6 summarizes the results obtained in the combat study by showing only the parameters which revealed a statistically significant difference from normal. It is evident that the Acute Combat Group, which had been exposed to intensive action for 18 hr, exhibited a markedly different metabolic pattern from the Prolonged

Combat Group, which had fought a less intensive action but for 5 days in the identical position occupied by the Acute Group. We interpreted the Acute Combat pattern as being typical of markedly increased adrenal cortical activity, whereas the Prolonged Combat pattern indicated the beginnings of adrenal cortical exhaustion; yet in both groups the blood patterns were little different from normal.

Let us turn now to the final body of data to be presented. Fig. 7 represents an attempt to summarize and compare the results of 5 major studies our laboratory has made, each involving a group of individuals exposed to stressful environments in real life situations for approximately the same length of time; *i.e.*, 12 to 36 hr. The Acute Combat Group from Korea is readily recognized. In addition there are shown the data from an international group of swimmers in the 1954 English Channel race, from a group of physiology graduate students and faculty exposed to high altitude at the University of California's White Mountain Research Station at 12,500 ft, from a group of U. S. Marine Corps personnel undergoing cold weather training at Pickles Meadow in the Sierra Nevada of California during winter, and from a group of U. S. Navy Seabees engaged in the pioneer construction of the McMurdo Sound Base in Antarctica during Operation Deepfreeze I.

No attempt will be made to interpret the figure in detail. The purpose in presenting it is to illustrate the kinds of differences that may be seen even in the present limited pattern approach. We are actively engaged at present in making detailed analyses of the physiological significance of these findings, and have not yet reached final conclusions. It is becoming evident, however, that different environmental stressors do produce metabolic responses, detectable by appropriate urine analyses, that vary in kind and degree. It also appears that the same stressor; namely, high altitude exposure at White Mountain, which is the only situation on which we have a repeat experiment, produces a uniform, reproducible response.

Summary

In summary, then, the general approach described here in much abridged fashion of determining patterns of metabolic response to environmental stressors offers a new means of evaluating the physiologic cost of adaptation to a particular stressor. Perhaps of more fundamental importance, it also affords the physiologist another means by which he can unravel the complex of interlocking mechanisms usually called the life process.

I cannot close without projecting future plans. Obviously, a great deal must yet be done to validate completely some of the ideas set forth. More real life situations involving different types of stressors must be examined. Controlled experiments under laboratory conditions need to be carried out wherein individual variables can be isolated. Finally, one of the most intriguing aspects of all involves the extension of this approach to the realm of pure psychic stressors and their physiological resultants. The effects of anxiety, of fear, of sustained anger all need to be examined in this light. Psychosomatic effects are well recognized but poorly understood, yet it is clear that such common and diverse ailments as ulcers, asthma, hypertension and ulcerative colitis, to name but a few, have common physiological links.

By understanding completely the physiology of man in relation to environment, we can precisely define optimal environments, so that the engineer can design and construct means whereby they may be attained.

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DISCUSSION

F. K. HICK, M.D., Chicago, Illinois: Reading this paper was very enjoyable, and my discussion was—"Do not expect to get from the physiologist soon by this technique a statement of what the comfort chart ought to be." That sounds very innocuous.

In hearing the paper presented, however, it seems best to defend this statement which is really an opinion. Why do I say this? There are two experiences which lead me to speak thus.

One is that we have been hunting for about 10 years for physiologic variables which show that sick patients are actually under stress when they are in a hot enough environment to make them sweat freely but not to give them a fever. We have not been able to find any, and under those conditions have not observed increases in 17-hydroxysteroid output.

The other experience has been in checking the comfort zone in aged patients, working with subjects between 50 and 85 years of age. We find that essentially the comfort zone is as given in the familiar experience tables of the Society; the explanation being very simply that these people dressed to suit themselves. It is as simple as that. We are not testing their response, we are testing the skill with which they have learned to clothe themselves.

How one can measure stress of such a small degree that it can be adjusted by mere changes of clothing which become habitual patterns for the man, is beyond my current imagination.

AUTHOR'S CLOSURE: I agree with Dr. Hick that it is not likely the 17-hydroxycorticoids *per se* are necessarily a good measure of all forms of stress. In fact, it is evident that no one body metabolite can serve as a universal index, and that is precisely this reason for pursuing the concept of determination of a biochemical profile. A good analogy can be made with the effective temperature index, which is comprised not of one variable but of several, measured simultaneously.

In regard to the sensitivity of physiologic measures, it must be remembered that we are merely at the threshold of correlation of stressful experiences with changes in body

chemistry. However, as more and more of the metabolic interplay is uncovered, the author is sure that the more subtle stresses will be found to be associated with measurable changes.

Nevertheless, Dr. Hick is correct in his final statement that if a stressor such as cold is eliminated by putting on more clothing or by turning up the furnace, there is little stress for the physiologist to measure. The difficulty here lies in the definition of comfort zone with respect to the true ambient environment. A Himalayan mountaineer suitably clad in his down-filled suit, his balaclava, his parka, his goggles, his fur-lined mittens, his vapor barrier boots and his portable oxygen supply is quite comfortable. But let him shed these and scarcely anyone will argue that the summit of a 25,000 foot peak is within the comfort zone.

1655

No. 1655

ENVIRONMENT REACTIONS IN THE 80 TO 105 F ZONE

By B. H. JENNINGS* AND BARUCH GIVONI**, CLEVELAND, OHIO

This paper is the result of research carried out by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Avenue, Cleveland 3, Ohio

THE COMFORT chart promulgated by ASHAE and based largely on research conducted before 1940 has served since its inception as a basic criterion for setting design and performance characteristics of installations for human occupancy. However, later research relative to environment, carried on by the Society and its cooperating institutions, and by numerous other organizations here and abroad has in some instances pointed to uncertainties in the comfort chart and to the resultant interpretations which follow from its teachings.

With a realization that factual certainty alone would suffice, the Committee on Research of the Society as far back as 1950 started planning a comprehensive program which would lead first, to a re-evaluation of all known information relative to the effects of environment and second, to the construction of a flexible facility which would permit the determination of new data, where required, and the re-evaluation of other data which appeared to need further scrutiny.

Study of the state of knowledge of human body behavior relative to environment, although never ending, was brought up to date and carefully considered before planning the new research facility. Design of the environment research facility was started in 1956, and construction started in 1957. Construction was sufficiently advanced by the summer of 1958 that testing with subjects could be started.

A description of the new environment laboratory (Fig. 1) is given here only in the briefest manner as a detailed description is being planned as the subject of a separate report. Let it here suffice to say that a study of the floor plan, Fig. 2, shows 2 pretest rooms, an observation (control) room, and finally the Environment

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Presented at the 65th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, January 1959.

Laboratory itself in which appropriate tests are conducted, using human subjects. The Environment Room is 12 ft wide, 24½ ft long, and the ceiling height can be adjusted from a low setting of 6 ft to a high setting of 11 ft. All interior surfaces are made of metal panels, on the back of which copper tubes are fastened, through which heated or chilled liquid can be circulated to provide controlled surface temperatures. The liquid circuits are so arranged that different temperatures can be

FIG. 1. . . .OUTSIDE VIEW OF THE ASHAE ENVIRONMENT LABORATORY SHOWING HUMIDIFICATION AND DEHUMIDIFICATION EQUIPMENT

maintained on different surfaces to simulate various room conditions, or as was true in the tests which are described in the paper, the wall temperatures were adjusted to produce surface temperatures essentially equal to the dry-bulb temperature being maintained in the room. The air-conditioning system, which is provided, can maintain air temperatures from 40 to 140 F dry-bulb and can control relative humidity at desired levels throughout this range. Air is introduced into the room through perforated inlet strips located between the ceiling panels and is exhausted from a continuous slot around the bottom periphery of the room. Provision is made for circulating up to 50 air changes per hour. An elaborate system of electronic-pneumatic control equipment has been provided to maintain and indicate test conditions. The space can be completely controlled from outside the test room so that minimum entry and exit is required by those in attendance.

The Environment Laboratory has come into existence as a result of the planned thinking of a large number of people, including members of interested Technical Advisory Committees, the Committee on Research and the Laboratory staff. The actual construction and detailed planning of the facility was under the direct supervision of R. G. Huebscher, with able assistance from C. M. Humphreys and L. F. Schutrum, all members of the Laboratory staff.

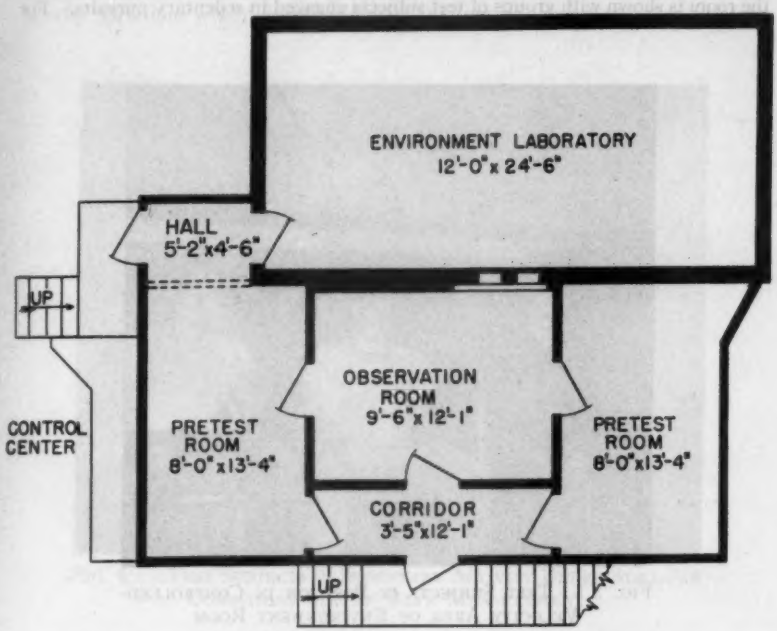

FIG. 2. . . . .FLOOR PLAN OF THE ASHAE ENVIRONMENT LABORATORY

SCOPE OF PRESENT RESEARCH

A precise re-evaluation of the comfort chart requires a determination of responses in the fringe areas of comfort, both in the warm and the cool regions. A decision was made that the first research program would be concerned with the warm region above and adjacent to the comfort zone. This decision was partially prompted by the fact that in prior research there appears to be a number of gaps relative to the performance of subjects under the environmental conditions considered here. Ultimately, the knowledge gained from investigations in this region will be extended to determine how patterns here merge into the comfort range.

The particular program described in this paper thus covers research work carried out in the approximate range of 80 F to 105 F dry-bulb temperature in an environ-

ment having uniform wall characteristics with the subjects first under conditions of still air, and secondly under conditions of air moving with a velocity of 120 fpm. The test work was carried out in 2 narrow relative-humidity bands, the lower at 30 to 33 percent, and the higher at 82 to 85 percent.

EXPERIMENTAL SETUP

The environment test room is the large area shown in Fig. 2. In Fig. 3 and Fig. 4 the room is shown with groups of test subjects engaged in sedentary pursuits. For

FIG. 3. . . .TEST SUBJECTS IN POSITION IN CONTROLLED-VELOCITY AREA OF ENVIRONMENT ROOM

the tests involving air motion, a fan was set up at the far end of the test room and was provided with a hooded entrance with arrangements carefully made so that a uniform pattern of horizontal air motion took place around the subjects. In most of the still-air tests, the fan was in position but inoperative and the only air motion was that associated with the natural ventilation of the room. Although the system air capacity is sufficient to handle up to 50 air changes per hour, because of the diffuse air distribution from the large ceiling area computation shows that the directed velocity past the subjects is less than 10 fpm at maximum flow. Tests with an Alnor thermoanemometer confirmed the fact that the air-motion values were exceptionally low at every location in the room. Thus, for the still-air tests it can be stated that air motion except for the local convective currents produced by the subjects themselves did not exceed 20 fpm. In the literature, air of this type is conventionally called still air.

For the moving-air tests, the draw-through fan of the wind tunnel was so adjusted that a uniform current of air at 120 fpm was drawn over and around the seated subjects. Baffles were installed and carefully adjusted to produce uniform air flow over the whole area. In addition to the baffling, the wind tunnel was provided with 3 layers of flyscreen in series. These were found to be effective in tempering velocity pulsations and providing greater uniformity in the flow pattern. The 120 fpm mentioned represented an adjusted average, as slight changes in fan speed from voltage variation, and unusual turbulence patterns caused occasional but rare velocity variations which dipped as low as 105 fpm to values as high as 135 fpm.

FIG. 4. . . .TEST SUBJECTS IN SEDENTARY ACTIVITY UNDER STILL-AIR CONDITIONS—PULSE RATE MEASUREMENT

The fan used was an axial-flow, propeller-type fan. The measurements of the air velocity in the wind tunnel passage were made with a calibrated thermoanemometer with checks also made by kata thermometers. In terms of these primary devices, a velometer was then calibrated and used to give quick reference readings of the air patterns maintained during the flow.

It has been mentioned that efforts were made to bring the radiant-surface and dry-bulb temperature into equality in the room. The success in this was relatively good, with differences of 1 F deg, or less existing between the globe-thermometer readings and dry-bulb temperatures. Because of this essential equivalence, dry-bulb temperature and globe temperature were considered interchangeable in these analyses although specific dry-bulb temperatures were used in making all graphs.

The relative humidity was measured by an aspirated psychrometer placed directly in the test chamber and shielded from radiant effects, and the control system was actuated by dry- and wet-bulb elements installed in the same device.

SUBJECTS AND PROCEDURE

For the subjective testing, 2 groups of test subjects were employed. The first group consisted of 3 or 4 young men in good health and ranging in age from 18 to 25 years. Most of the work was done with this group; however, to confirm the accuracy of testing with this group, a second test group was used in the early fall of 1958, which consisted of 4 women and 2 men, ranging in age from 18 to 22 years. The responses of the second group were in remarkably close agreement with the indications given by the first group.

This first 1958 testing program was carried out during summer and the subjects all wore light clothing. However, as it was desired to have the effect of clothing variations at a minimum throughout the tests, each subject wore exactly the same clothing in each test. For this purpose, when he arrived at the Laboratory in the morning, he was aked to sit quietly and rest for a period of from ½ to ¾ hr in a normal environment in the range of 74–78 F, with relative humidity ranging between 40 and 60 percent. With the subjects fully relaxed at the end of the rest period,

TABLE 1. . . . REFERENCE SCALES USED IN RECORDING SUBJECTIVE RESPONSES TO TEMPERATURE AND HUMIDITY

THERMAL SENSATION[1,2]	HUMIDITY SENSATION[2]
1—cold	1—wet
2—cool	2—very humid
3—slightly cool	3—humid
4—comfortable	4—normal
5—slightly warm	5—dry
6—warm	6—very dry
7—hot	7—painfully dry

their pulse rates and oral temperatures were measured and recorded. The laboratory clothing used by each subject was then weighed and the subject changed into these clothes. The subjects then entered the environment test room and first of all were asked to record their initial impression. They were then weighed on a balance sensitive to one-hundredth pound and then took their sitting positions. The same positions were occupied both during still-air testing and air-motion testing. The subjects were allowed, during the period of the tests, to pursue sedentary occupations such as reading, studying, or playing cards, if they wished; the only requirements being that they not move appreciably from their locations and that they not walk around except for that minimum activity which was required when they arose for intermediate weighings and to record impressions on their individual data sheets at half-hour intervals. Also on half-hour intervals, the test monitor measured the pulse rate of each individual and also measured oral temperatures. Each complete test lasted for 3 hr, with 2 tests per day normally being carried out with the same subjects. At the end of the 3-hr period, each subject was weighed in his laboratory clothes just as at the start of the test. Following this, the subject then

[1] Exponent numerals refer to References.

changed into his street clothes, and his test clothes were again weighed. This procedure made it possible to determine and correct for additional moisture absorbed in the laboratory clothing. At the higher relative humidities and temperatures in the range of 4 on the Sensible Perspiration Scale, it was found this involved corrections which approached 100 millipounds per hour per square meter of body surface. However, the corrections for numbers less than 2 on the Sensible Perspiration Scale were found to be so small (1 to 6 mlb/hr m^2) that in testing with the second group, it was not deemed necessary to correct for moisture absorption in the light clothes worn by the subjects. In general, the clothing worn by the subjects during the tests consisted of a sport-type shirt with short sleeves, light underpants, socks, and shorts or skirts, as the case might be. The clothes, in general, without shoes, weighed from 1½ to 2 lb.

TABLE 2. . . .REFERENCE SCALE USED IN RECORDING SUBJECTIVE RESPONSE TO SENSIBLE PERSPIRATION

0—forehead or body, dry
1—forehead or body, clammy (surface feels moist to touch)
2—forehead or body, damp (perspiration just visible)
3—forehead or body, wet (sweat covering the surface, frequently in drops)
4—parts of the clothing—wet
5—most of all of the clothing—wet

TABLE 3. . . .REFERENCE SCALES USED IN RECORDING SUBJECTIVE RESPONSES TO AIR MOTION AND PLEASANTNESS

AIR MOTION SCALE	PLEASANTNESS SCALE
0—Stagnant	1—pleasant
1—still	2—indifferent
3—just perceivable	3—unpleasant
4—medium wind (air motion)	4—very unpleasant
5—strong wind (air motion)	

The reference scales which the subjects were asked to use in recording their impressions were purposely selected to be in agreement with scales which had been employed by a number of prior researchers in the field and are listed in Tables 1 to 3[1, 2, 3].

The humidity sensation scale, however, was found to have almost no definitive value and it was soon decided that it was not worth recording. Subjects appeared to have almost no ability to note changes in relative humidity. Even after instruction as to what phenomena one might except to associate with humidity variations of the air, only haphazard agreement was found in these tests.

The scale for *sensible perspiration*[1], which was very significant, is indicated in Table 2.

The *scale of air motion*[4] and the *scale of pleasantness* were also employed in some cases; however, these were not particularly significant for these tests. For moving air the values named fell in the category of *slight breeze* or *just perceivable*. As

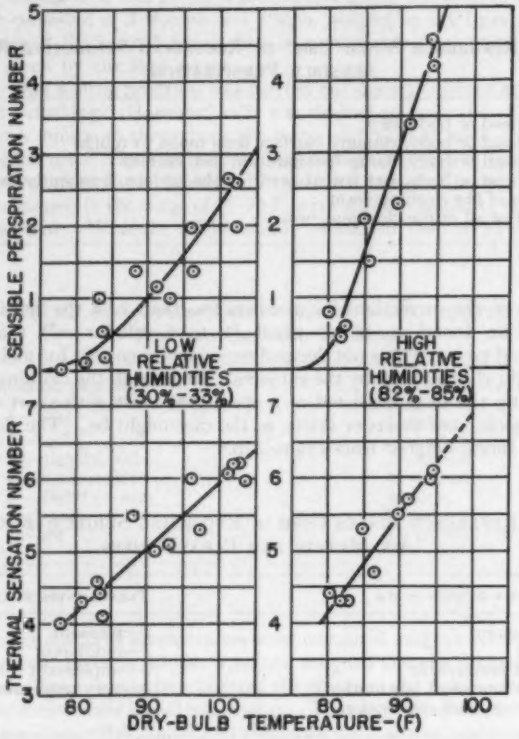

FIG. 5. . . .IMPRESSION RESPONSES OF SUBJECTS TO STILL-AIR CONDITIONS UNDER LOW AND HIGH RELATIVE HUMIDITIES

far as pleasantness is concerned, the reactions were either *unpleasant* or *very unpleasant* and showed little significance.

SUMMARIZED DATA

As could be expected, the amount of data taken in experiments of this type is extremely voluminous. Consequently, it was decided to present the data in the form of a series of graphs. The data obtained are of 2 types *viz:* (1) the subjective re-

sponses and (2) the physiological responses. At this point, mention should be made of a phenomenon which was very much in evidence; namely, that the responses of the subjects, both subjective and physiological, were different in morning and afternoon tests. The most striking differences in this connection occurred in relation to the physiological responses where the variation between a morning test reaction and an afternoon test reaction at times was greater than the variation existing between test results for still air and moving air. This characteristic of

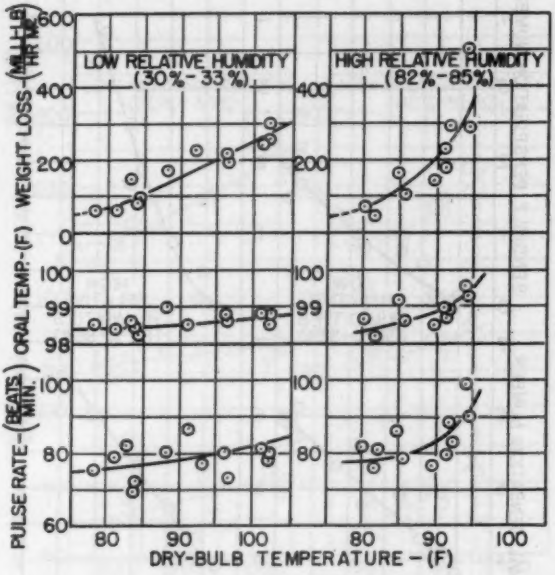

FIG. 6. . . .PHYSIOLOGICAL RESPONSES OF SUBJECTS TO STILL-AIR CONDITIONS AT LOW AND HIGH RELATIVE HUMIDITIES

human behavior is not new, as it has been discussed in the literature in several places, one of the most recent and complete studies being that of Iampietro et al[8]. Because separate morning and afternoon analyses would needlessly complicate the problem, daily averages were used.

Still Air: In Fig. 5 is shown the pattern of subject responses in still air for the 2 ranges of relative humidity. Below 89 F, at low relative humidity, the subjects indicated an impression of being between comfortable and slightly warm, *i.e.*, less than 5 on the scale. At the higher relative humidity, the sensation number exceeded 5 at a lower temperature, approximately 86 F, and as would be expected, the thermal sensation curve had a much steeper slope. The sensible perspiration

response is more drastic, at high relative humidity, reaching a value of 3 at 90 F, whereas at the lower relative humidity, the value of 3 is not reached until almost 105 F.

Physiological responses for the subjects are shown in Fig. 6. At low relative humidity, below 85 F, the skin of the subjects was dry, the pulse rate approximately

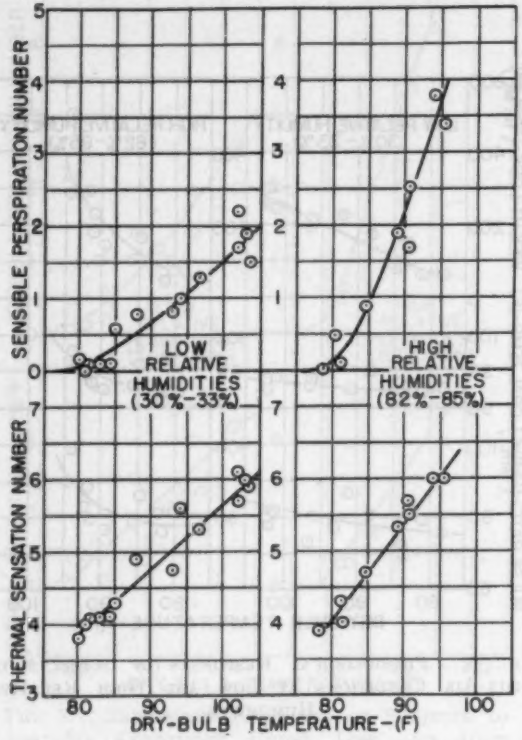

FIG. 7. . . .IMPRESSION OF SUBJECTS AT LOW AND HIGH RELATIVE HUMIDITIES UNDER AN AIR MOVEMENT OF 120 FPM

76 beats per minute, and oral temperatures in the normal range. The sweat rate was also modest, below 100 milli-pounds* per hour per sq meter (mlb/hr m^2) of computed body surface (45 grams per hour per square meter). With increasing air temperature, the rise in all the responses is gradual and essentially linear. At

* 1000 millipounds = 454 grams = 16 oz; 1 gram = 2.2 millipounds = 0.035 oz.

the higher relative humidity, the physiological responses show a more abrupt pattern. With increasing temperature the stress on the body, as indicated by sweat production†, rises very rapidly. The rapid rise in pulse rate and oral temperature commences when the temperature rises above 90 F.

A change in relative humidity from a low to high value reduces the evaporative capacity of the air in contact with the body. In an effort to maintain the effectiveness of evaporative cooling, with increasing temperature, the body increases the wetness of the skin or at least the wetted area. This is shown in Fig. 5 for the sub-

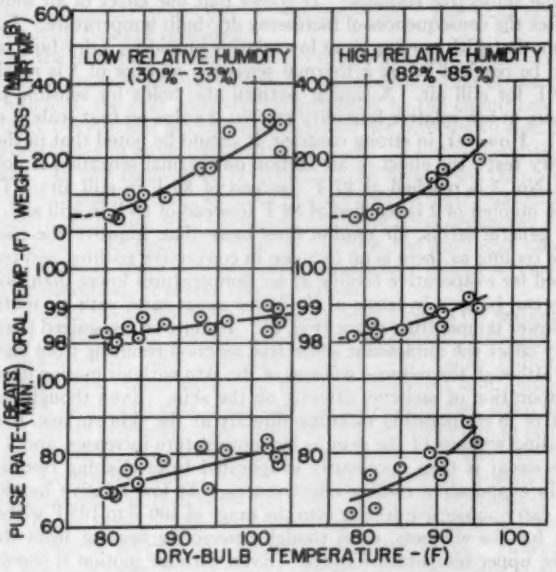

FIG. 8. . . .PHYSIOLOGICAL RESPONSES OF SUBJECTS AT LOW AND HIGH RELATIVE HUMIDITIES UNDER AN AIR MOVEMENT OF 120 FPM

jects with the sensible perspiration index rising to the moist No. 2 value on the perspiration scale. The greater wetness of the skin promotes greater evaporation of the sweat, and up to a point, all of the sweat is evaporated. However, the increase in wetness of the skin indicates that a larger amount of sweat is probably evaporated on the clothing instead of directly on the skin. Evaporation from the clothing necessarily reduces the efficiency of evaporative cooling when the clothing is loose and not always in contact with the skin. At air temperatures rising above

† More accurately the term which should be used here is *evaporative weight loss*.

90 F, at high relative humidity, the differential between the vapor pressure at the skin surface and vapor pressure in the air may not be sufficient to produce adequate evaporation to offset the body heat load. Hence, although there is a sharp rise in sweat production, the sweat cannot be evaporated and only reflects the unsuccessful effort of the body to maintain normal thermal balance. This is also indicated by the steeper rise in pulse rate and oral temperature.

The increase in wetness of the skin under adverse environmental conditions is usually associated with a feeling of discomfort.

Moving Air (120 fpm): The test results involving moving air are illustrated in Figs. 7 and 8 and can be compared directly with Figs. 5 and 6 for still air. Fig. 7 delineates the subjective response. It shows that one effect of air motion is partially to offset the consequences of increasing dry-bulb temperature. For example, it can be seen that with moving air at low relative humidity, a dry-bulb temperature of 93 F can be reached before a thermal sensation number of 5 is reached in contrast to 90 F for still air. A similar pattern also holds for sensible perspiration number where at low relative humidity the No. 2 value on that scale is not reached until 105 F. However, in strong contrast, it should be noted that in the high relative-humidity tests, the effect of air motion on thermal sensation is not as significant and a No. 5 is reached at 87 F (instead of 86 F in still air). The sensible perspiration number of 2 is reached at 89 F (instead of 86 F in still air).

In more general terms, air motion does more than improve the mechanism of evaporative cooling as there is an increase in convective cooling, and consequently reduced need for evaporative cooling at air temperatures lower than skin temperature. This can be seen in terms of the lower sweat rates with air motion (vs still air) in the lower temperature range (Fig. 8). It must be recognized that air motion cannot fully offset the unpleasant subjective reaction resulting from higher relative humidities although the reduced wetness of the skin with air motion does arise from greater evaporation of moisture directly on the skin. Even though moving air is more effective in evaporating moisture directly at the skin surface, it cannot prevent increasing wetness of the skin as air temperature increases, and a larger fraction of the sweat is then necessarily evaporated from clothing resulting in some reduction in evaporative cooling effectiveness. At low relative humidity, it was possible to carry subjective testing into the range of 100 F to 105 F without extreme discomfort for the subjects, even though convective heating from the air could occur at the upper temperature range. Even with air motion it should be noted, however, that there is a gradual rise in pulse rate and oral temperature both at low and high relative humidity.

In summary, air motion tends to diminish the magnitude of thermal stress which exists for the body in a warm environment. At low humidity, the evidence of this does not show up to a great extent as reduced sweat rate. However, at high humidity there is an appreciable reduction in sweat rate as the air motion enables the body to use the sweat more effectively in its cooling process.

Still air and air at 120 fpm were compared in these tests. However, several check tests were carried out employing air at other velocities, and an analysis of these tests compared with 120 fpm air indicated that increasing improvement did occur with air velocities up to 120 fpm, but the additional improvement with velocities increased to 200 and 300 fpm was small in amount. That is, it is extremely desirable for air to move past the body with sufficient linear motion to remove both evaporated moisture and some convective heat, but no great improvement occurs when the air movement is at much higher velocities; and at air temperatures above skin temperature convective body heating can be a factor.

ANALYSIS OF RESPONSE FACTORS

One of the ultimate objectives of the program of testing being carried out is to find, if possible, a set of response factors which would be indicative for the whole range of environmental conditions. This is a difficult assignment, and in this investigation only a limited range of environment has been covered and has shown how certain subjective responses and physiological reactions to the various conditions are related. In this paper, no mention has been made of the various comfort indexes which have already been employed such as effective temperature[1, 6, 7, 8], operative temperature[9, 10, 11], equivalent temperature[12, 13], the predicted 4-hr sweat rate[14], and the heat-stress index[15]. Some of these are based on physiological responses, others on purely subjective or comfort votes. The data here presented have included both types of responses.

A survey of the tests here presented shows that oral temperatures and pulse rate appear to have extremely limited usefulness. An examination of the graphs indicates that although the oral temperature increased as the conditions of thermal stress increased, the rise was gradual and certainly not sufficiently significant to act as an important index of environmental conditions. A similar condition holds for the pulse rate. In this connection, it should also be mentioned that pulse rates in individuals are extremely erratic, as it can happen that an individual pulse rate may move completely out of line with environmental conditions depending on what the subject is thinking or his emotional state at the time the reading is taken. Thus, it is felt that neither of these parameters has significance except at very high temperatures or at conditions of severe thermal stress.

It is felt that more confidence can be placed in weight loss in the warm region than in the other 2 physiological measurements just considered, although even with weight-loss measurement, wide variability exists. The values indicated in the graphs for weight loss (sweat rates) are relatively constant and uniform. However, it should be mentioned that in each subject the weight loss, when measured at ½ hr intervals, shows extremely wide variations. For weight loss to be representative of the environment, it appears necessary to totalize the loss over a period of time, probably for not less than 2 hours. In this connection, attention should be called to the consistency of results for the tests carried out in connection with the so-called 4-hr sweat rate[14]. This was found to be very useful for environment analysis in the case of severe (hot) environmental conditions. However, as the environment becomes more comfortable and thermal equilibrium is approached, the magnitude of the weight loss becomes small and difficult to measure.

Subjective feelings appear to be very important in the warm region, as the evidence here points to the fact that the subject himself is a fairly good judge of his relationship to environment. Wide variability in indicated results does exist, but the thermal sensation number and the sensible perspiration number (wetness of the skin) were found to be sensitive and reproducible responses. In this region, the responses must be used together as the cause of discomfort may be high temperature or wetness of the skin and clothing or combinations of these. For example, an individual can be cool but very uncomfortable if the relative humidity is high and the air is still even though the dry-bulb temperature of the air is relatively low. People sometimes associate the appearance of sweat with some feeling of cooling, but this is not necessarily the case and a subject can be far from comfort under a condition of visible perspiration. One basic difference between these responses is that the thermal sensation is almost instantaneous and does not change appreciably with time of exposure in a constant environment. On the other hand, the

sensible perspiration in a warm environment is a function of time of exposure. It thus appears desirable to consider for analysis the use of the subjective responses along with measures of sweat rates, particularly as the higher limits of the temperature range are reached.

ACKNOWLEDGMENTS

The help of other members of the staff during the conduct of this research is acknowledged with sincere thanks. Particular thanks are extended to Walter Koch, M.D., who, although he joined the Laboratory staff when this project was almost completed, was most helpful in giving advice and assisting in the analysis of the results.

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DISCUSSION

R. C. NEVINS, Manhattan, Kansas (WRITTEN): It is gratifying to see the beginning of the Society's program to revaluate the comfort chart. The data presented are, it is assumed, only a limited beginning of an extensive program. Since the test group was limited in size, the results must be interpreted with caution.

Two questions come to mind regarding the presentation of the data.

1. What methods were used to average the subjective data to arrive at a value for plotting on the various graphs?
2. Were the first impressions included in these averages particularly with reference to Thermal Sensation values?

With regard to the thermal sensations for the subjects exposed to still air conditions, a comparison of the data presented by Jennings and Givoni with data obtained at Kansas State College* shows the following. For similar conditions of low humidities (30–60 percent) and still air conditions, the average sensation number of 39 male college students when exposed for one hour to an air temperature of 75 F and surface temperatures of 75 F was 3.9 with a standard deviation of 0.387. Jennings and Givoni have obtained a value of 3.7 or 3.75. For 80 F air temperature and 80 F surface temperatures Kansas State College data show an average value of 4.2 for 7 students compared with the Laboratory value of 4.2 and for 85 F air temperature and 85 F surface temperatures, Kansas State data show an average value of 5.13 for 36 subjects while the Laboratory data show a value of 4.6. The latter discrepancy may be due to a slightly higher air motion at Kansas State although no values greater than 25 fpm below the 4 ft level were measured.

The limited usefulness of oral temperature and pulse rate is to be expected due to the variations which exist between subjects due not only to physiological considerations but also to psychological stresses.

CHARLES S. LEOPOLD, Philadelphia, Penn., (WRITTEN): The results shown in Fig. 5 of this paper are in radical disagreement with all previous Laboratory work on this subject, as illustrated by Fig. A which indicates the 1923 chart, the modification as shown in THE GUIDE 1958, and the result of the present investigation. This present paper states that "... the thermal sensation is almost instantaneous and does not change appreciably with time of exposure ...". This observation casts doubt on one criticism frequently directed at previous work. One must conclude either that the previous Laboratory work and this report do not cover the same variables, or that the early or the late technique is in error, or that people have changed.

In the first category there are the factors of clothing, a limited age group—which previously did not appear significant, and the fact that odors were substantially absent due to the use of carbon filters, not mentioned in the text.

The odor factor should not be dismissed without further study. On one project, and only one, with unusually good odor removal facilities, both by liquid sorbents and charcoal, the first indication of high humidity in a room at approximately 76 F was water dripping from insulated pipes rather than the subjective reaction of the occupants. It was in view of this observation that the discusser recommended that charcoal filters be included.

In the earlier experiments the mean radiant temperature was not maintained quite so close to the room air temperature as in these experiments but it is unlikely that this would account for any substantial part of the discrepancy in the lower temperature range.

The attempt to separate sensations of warmth from sensations of moisture does not seem valid in determining comfort and may lead to a biased approach on the part of the subject.

Sir John Parsons, noted British investigator in the field of illumination, stated: "It is the custom, incontrovertibly justified by pragmatic sanction, for the physicist to attack

* R. G. Nevins and A. O. Flinner: Effect of heated-floor temperatures on comfort (ASHAE TRANSACTIONS, Vol. 64, 1958, p. 175).

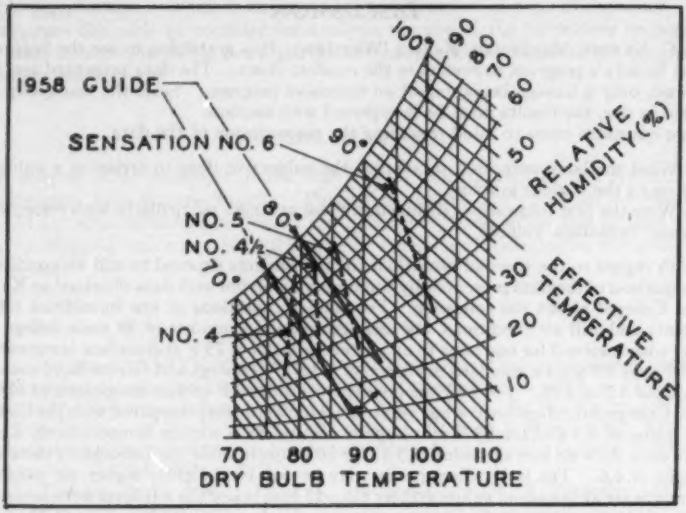

FIG. A. . . .COMPARISON OF DATA FROM ASHVE RESEARCH PAPER NO. 655 (1923) WITH DATA FROM JENNINGS AND GIVONI PAPER AND WITH DATA IN THE GUIDE 1958

his problems by a process of analysis and simplification. Hence it behooves him to remember what the biologist is less likely to overlook, namely, that animal behavior is the response to the situation as a whole, which can in no case be split up into fragments without profoundly modifying both the whole and the parts."

As regards the possibility of error in this series of experiments, the reader of the paper is at a disadvantage because the authors have not included the actual data obtained to show the scatter of the individual tests, as was done in reporting some of the previous investigations.† For example, one should like to know whether, on Fig. 5, at low humidity, *Thermal Sensation 4*, there are enough No. 3 votes to justify the conclusion and whether there was evidence based on the complete data for the extrapolation of the high humidity line to Sensation 4.

The paper states that there was a significant discrepancy between morning and afternoon experiments. Was there any justification for averaging results other than the limited amount of data available? A major difference could be of engineering significance.

As a former chairman of the committee, having been somewhat instrumental in inaugurating this program, your present discusser argued the advantages of having a first-class psychologist assigned to this work and stated that he considered the medical aspects relatively unimportant in the comfort region, also a slight bias toward interpreting tolerance as unqualified preference could lead to large error. It is again suggested that

† F. C. Houghten, F. E. Giesecke, Cyril Tasker, and Carl Gutberlet: ASHVE RESEARCH REPORT NO. 1055—Cooling requirements for summer comfort air conditioning (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 145).

some consideration be given to adding a professional psychologist to the team before attempting further work in the Laboratory.

Because of the authority implied in the work of our Laboratory, it is thought that this paper should not appear in the TRANSACTIONS without detailed data and a statement of the method of analysis.

G. ROZEN, Mt. Vernon, N. Y.: This discusser also compared the results of this paper with the effective temperature data in a manner similar to that shown by Mr. Leopold on Fig. A., and agrees with many of Mr. Leopold's remarks.

The discusser indicates further that at the low temperature range of this experiment, 80 F, the line of equal thermal sensation has a slope in the reverse direction from that of the effective temperature lines. At the high temperature, at least, the slope is approximately in the same direction as the effective temperature lines.

In comparing the air velocity experiment, with that of the still air experiment, one observes that at air temperatures lower than skin temperature, the higher the velocity the better is the sensation of warmth. That is evidently true. With air temperatures higher than skin temperature, again the higher the velocity the better sensation of warmth was reported. That is a discrepancy concerning which it would be helpful to have the authors' discussion.

DR. THOMAS BEDFORD, London, England: Having seen this paper in the typed stage it seemed that it would be worthwhile to compare the results described by the authors with the results of this discusser's earlier work of more than 20 years ago, insofar as comparison is possible.

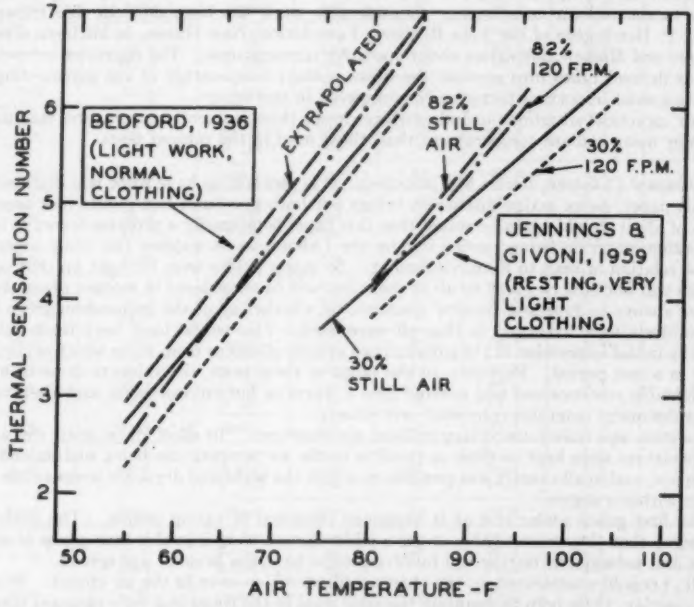

FIG. B. . . .COMPARISON OF RESULTS OF WORK OF BEDFORD (1936) AND DATA PRESENTED BY JENNINGS AND GIVONI

The discusser had an equation that described the comfort votes of his subjects in terms of the different environmental variables, and from it computed comfort votes for various environments used by the authors of this paper. The results of those calculations are shown on the left of Fig. B, while the results reported by the authors are shown on the right of the diagram. There is a certain overall agreement except in the position of the curves with respect to air temperature My subjects were doing light jobs in industry. They were wearing their normal British-type clothing, which is heavier than the American subjects wore, and they were also accustomed to living in distinctly cooler conditions than were these American subjects. These facts explain the displacement to the left, and otherwise there is a very considerable similarity in scatter and in the slope of the curves. It would seem there is a very considerable agreement between the results of the two investigations, and that the differences can be explained away.

PETER FLACK, New York, N. Y.: Here is a question for Professor Jennings on whether he intends to change the subjects to an age group of 25 to 45 years which would seem to be more representative than the present test group aged 20 to 25 years.

Also, it is felt that while the clothing worn by the girls would be perhaps normal, the men in most instances would be wearing jackets, or else, have rolled down shirt sleeves.

It seems certain that age and clothes as mentioned in Dr. Bedford's paper, would affect the results.

During the temperature tests in the range of 80 to 105 F described in this paper, the mean radiant temperature was kept at ambient temperature. It would be helpful to know whether the authors intend to vary the mean radiant temperatures of the surrounding surfaces.

The effective temperature previously was used by ASHAE without any weight being given to the radiant component. Considerable work has been done in this respect. Dr. L. P. Herrington of the John B. Pierce Foundation, New Haven, in his book, *Temperature and Human Life* writes about operative temperatures. The operative temperature as defined takes into account the mean radiant temperature of the surroundings. This is a most important factor to be considered in this study.

Most practical situations include surfaces lower than ambient temperature, resulting in lower mean radiant temperatures than those used in the present tests.

AUTHORS' CLOSURE, (Oral) MR JENNINGS: It is pleasant to have some real discussion of this paper, as an active discussion brings out both good and bad points in a paper. First of all, it should again be stated that this paper is primarily a progress report in the long-range program being carried out by the Laboratory to explore the many aspects of the relation of man to his environment. So many points were brought up that oral replies will not now be given to all of them but will be considered in written discussion.

The answer to Professor Nevins' question, on whether all of the impressions given by the subjects were averaged, is that all were used. This would have been inadvisable had the initial impression of the subject been greatly different from those which occurred later in a test period. However, in the range of these tests, the subjects came from a comfortable environment and entered into a warm or hot environment, and their first and subsequent reactions coincided very closely.

Question was raised concerning radiant temperatures. In all of these tests, the wall temperatures were kept as close as possible to the air temperature being maintained in the space, and in all cases it was possible to adjust the walls and dry-bulb temperature to range within a degree.

The first group under test as it happened consisted of young people. The authors are sorry that they were unable to get a wider spread of ages in this first group of subjects, but subsequent testing will involve groups having a broader age spread.

Mr. Leopold commented on the charcoal filters which were in the air circuit. Without question, these help to maintain the odor level in the room in a very pleasant condition. In fact, the authors were amazed at the change in odor level which occurred when, for a short period, the charcoal filters were not in service. Mr. Leopold's remark about

TABLE A. . . .REPRESENTATIVE TEST DATA SUMMARIZED

DATE BEFORE OR AFTER NOON AIR MOTION	DRY-BULB, F WET-BULB, F GLOBE, F R.H., %	SUBJECTS	WT LOSS, MILLI-POUNDS PER HR SQ M	PULSE RATE, PER MIN	ORAL TEMP, F	THERMAL SENSATION NUMBER	SENSIBLE PERSP. NUMBER
7/29/58	93.0	R (M)	312	77	98.6	5.0	1.2
a.m.	69.5	D (M)	250	76	98.6	5.1	0.8
	91.5	B (M)	323	78	98.2	5.1	1.0
Still	30%	mean	295[a]	77	98.4	5.1	1.0
8/27/58	81.0	G (M)	180	79	98.4	4.3	0.3
p.m.		D (M)	158	74	98.2	4.4	0.0
	60.9	B (M)	220	83	98.6	4.4	0.3
	81.9	R (M)	140	82	97.7	4.0	0.0
Still	30%	mean	175	79	98.2	4.3	0.2
8/1/58	96.0	R (M)	370	78	98.8	6.0	2.1
p.m.	72.0	D (M)	442	78	98.8	6.0	2.2
	96.7	B (M)	417	84	98.8	5.9	1.8
Still	31.1%	mean	410	80	98.8	6.0	2.0
8/8/58	101.0	G (M)	500	91	98.6	6.0	2.7
a.m.	76.1	D (M)	467	75	98.8	6.0	2.5
	101.5	B (M)	444	76	98.9	6.4	3.0
Still	32%	mean	470	81	98.8	6.1	2.7
8/15/58	103.3	G (M)	657	94	97.8	5.8	3.3
p.m.	77.0	D (M)	700	88	99.1	6.1	3.6
	104.0	R (M)	563	106	98.3	5.9	3.1
Still	30%	mean	640	97	98.7	5.9	3.3
9/9/58	103.6	G (M)	514	83	98.4	6.3	2.4
a.m.	76.8	D (M)	514	76	98.2	6.0	1.6
	102.0	B (M)	438	74	98.8	6.2	2.1
Still	30%	mean	489	78	98.5	6.2	2.0
9/9/58	103.4	G (M)	520	76	98.6	6.3	2.4
p.m.	76.9	D (M)	570	80	98.9	6.1	2.8
	102.0	B (M)	602	85	99.0	6.1	2.6
Still	30%	mean	564	80	98.8	6.2	2.6
9/11/58	91.3	G (M)	362	86	98.7	5.8	3.1
a.m.	85.8	D (M)	262	74	98.6	5.8	3.0
	91.3	B (M)	362	76	98.7	5.5	3.2
Still	82%	mean	328	79	98.7	5.7	3.1
9/11/58	91.7	G (M)	600	85	98.8	5.9	3.5
p.m.	85.9	D (M)	600	77	98.9	5.9	3.9
	91.8	B (M)	488	87	99.0	6.0	3.9
Still	81%	mean	569	83	98.9	5.9	3.8
9/2/59	89.5	G (M)	260	77	98.5	5.2	2.0
a.m.	85.3	D (M)	240	72	98.6	5.5	2.1
	90.3	B (M)	310	78	98.5	5.7	2.7
Still	84%	mean	270	76	98.5	5.4	2.3
8/21/58	91.0	G (M)	394	95	98.8	5.5	3.4
p.m.	85.8	D (M)	470	80	99.2	5.9	3.5
	91.5	R (M)	360	90	98.8	5.7	2.9
Still	82%	mean	408	88	99.0	5.7	3.3
8/22/58	81.5	G (M)	140	82	98.3	4.1	0.4
a.m.	77.0	D (M)	70	69	98.3	4.6	0.5
	81.2	R (M)	44	77	98.0	4.0	0.0
Still	81%	mean	85	76	98.2	4.2	0.3

TABLE A. . . . REPRESENTATIVE TEST DATA SUMMARIZED (*continued*)

DATE BEFORE OR AFTER NOON AIR MOTION	DRY-BULB, F WET-BULB, F GLOBE, F R.H., %	SUBJECTS	WT LOSS, MILLI-POUNDS PER HR SQ M	PULSE RATE, PER MIN	ORAL TEMP, F	THERMAL SENSATION NUMBER	SENSIBLE PERSP. NUMBER
8/22/58 p.m.	82.0 78.0 81.7	G (M) D (M) R (M)	194 250 162	81 80 86	98.8 98.9 98.2	4.1 4.6 4.1	0.4 0.8 0.1
Still	83%	mean	202[a]	82	98.6	4.3	0.4
8/28/58 a.m.	80 75.5 80.7	G (M) D (M) B (M) R (M)	170 160 110 50	86 81 81 80	98.6 98.8 98.6 98.7	4.5 4.5 4.5 4.1	0.8 1.0 1.0 0.4
Still	82%	mean	122	82	98.7	4.4	0.8
9/4/58 a.m.	85.5 81.0 86.5	G (M) D (M) B (M)	191 160 209	78 83 72	98.3 98.8 98.6	4.7 5.0 4.5	1.3 1.6 1.6
Still	83%	mean	194	78	98.6	4.7	1.5
9/5/58 a.m.	94.2 89.8 95.0	G (M) D (M) B (M)	430 468 740	91 85 92	99.1 99.0 99.7	6.1 6.1 6.2	4.0 4.0 4.6
Still	84%	mean	545	89	99.3	6.1	4.2
9/5/58 p.m.	94.0 89.0 95.2	G (M) D (M) B (M)	1000 948 875	98 96 104	99.3 99.5 100.0	6.0 6.0 6.0	4.6 4.6 4.6
Still	83%	mean	941	99	99.6	6.0	4.6
9/10/58 a.m.	103.5 76.4 102.0	G (M) D (M) B (M)	601 501 —	80 72 —	98.4 98.6 —	5.9 5.9 —	1.4 1.5 —
120 fpm	30%	mean	551	76	98.5	5.9	1.5
9/10/58 p.m.	103.2 76.4 103.0	G (M) D (M) B (M)	664 312 250	85 81 77	98.8 99.3 98.6	6.1 5.9 5.9	1.9 1.9 1.9
120 fpm	30%	mean	409	81	98.9	6.0	1.9
8/8/58 p.m.	102.0 76.5 101.5	G (M) D (M) B (M)	625 578 547	87 77 87	98.8 99.0 99.3	6.0 6.2 6.0	1.8 3.0 1.8
120 fpm	31%	mean	583	84	99.0	6.1	2.2
8/1/58 a.m.	94.0 70.5 95.0	R (M) D (M) B (M)	328 300 344	71 70 68	98.3 98.3 98.2	5.3 5.8 5.8	0.8 1.1 1.0
120 fpm	31.0%	mean	324	70	98.3	5.6	1.0
8/27/58 a.m.	81.0 60.9 81.0	G (M) D (M) B (M) R (M)	100 120 92 97	71 63 69 73	98.0 97.8 98.4 97.3	4.0 4.0 3.9 4.0	0 0 0 0
120 fpm	30%	mean	102	69	97.9	4.0	0
7/29/58 p.m.	93.0 69.5 91.0	R (M) D (M) B (M)	302 312 360	78 79 85	98.7 98.9 98.4	4.8 4.8 4.7	0.7 1.0 0.9
120 fpm	30%	mean	325	81	98.7	4.7	0.8
8/28/58 a.m.	78.5 74.5 79.5	G (M) D (M) B (M) R (M)	88 81 69 63	66 64 64 63	97.8 97.8 97.8 —	3.6 4.1 3.8 4.0	0 0 0 0
120 fpm	83%	mean	75	64	97.8	3.9	0

TABLE A. . . .REPRESENTATIVE TEST DATA SUMMARIZED (*concluded*)

DATE BEFORE OR AFTER NOON AIR MOTION	DRY-BULB, F WET-BULB, F GLOBE, F R.H., %	SUBJECTS	WT LOSS, MILLI-POUNDS PER HR SQ M	PULSE RATE, PER MIN	ORAL TEMP, F	THERMAL SENSATION NUMBER	SENSIBLE PERSP. NUMBER
8/29/58 a.m.	94.8	G (M)	110	79	98.2	4.9	0.9
	80.2	D (M)	60	68	98.2	5.0	1.2
	84.9	B (M)	110	79	98.6	4.4	1.1
		R (M)	40	77	—	4.7	0.3
120 fpm	83%	mean	80[a]	76	98.3	4.7	0.9
9/2/58 p.m.	89.0	G (M)	222	78	98.8	4.8	1.7
	84.5	D (M)	269	78	99.0	5.6	2.6
	90.0	B (M)	303	83	98.9	5.6	3.0
120 fpm	83%	mean	265	80	98.9	5.4	2.4
9/3/58 a.m.	95.3	G (M)	331	91	98.9	5.8	3.0
	90.3	D (M)	300	91	98.9	5.8	3.5
	95.0	B (M)	500	80	99.1	6.5	3.6
120 fpm	83%	mean	377	84	98.9	6.0	3.4
9/3/58 p.m.	94.0	G (M)	350	84	98.8	5.9	3.6
	89.5	D (M)	500	87	99.3	5.9	4.0
	95.5	B (M)	418	92	99.4	6.3	3.7
120 fpm	85%	mean	419	88	99.2	6.0	3.8
9/12/58 a.m.	90.6	G (M)	281	83	98.2	5.8	2.4
	86.0	D (M)	187	73	98.5	5.9	2.6
	91.1	B (M)	269	78	98.4	4.9	1.7
120 fpm	84%	mean	246	78	98.4	5.5	2.2
9/12/58 p.m.	90.2	G (M)	288	75	98.4	5.9	3.1
	87.0	D (M)	319	73	98.9	6.1	3.4
	90.3	B (M)	338	74	99.1	5.1	2.8
120 fpm	86%	mean	315	74	98.8	5.7	3.1
11/3/58 p.m.	78.1	C (F)	44	76	98.9	4.0	1.0
		J (F)	—	86	97.8	4.0	1.0
		F (F)	—	97	98.2	3.3	0.0
	59.5	S (F)	114	83	98.6	4.0	0.0
		G (M)	125	76	99.0	5.3	0.9
	78.9	T (M)	178	79	98.2	3.5	0.5
Still	31%	mean	115	83	98.5	4.0	0.6
11/11/58 a.m.	75.0	C (F)	23	65	98.6	3.9	1.0
		—	—	—	—	—	—
		F (F)	21	69	98.0	4.2	0.0
	71.5	S (F)	10	75	98.7	4.0	1.0
		G (M)	31	72	98.1	3.6	0.3
	75.7	T (M)	80	79	97.9	3.0	0.0
Still	85%	mean	33[a]	72	98.3	3.8	0.5
11/11/58 p.m.	75.1	C (F)	48	74	98.7	4.0	1.0
		J (F)	—	—	—	—	—
		F (F)	—	67	98.1	4.0	0.0
	71.8	S (F)	115	77	98.4	4.0	0.0
		G (M)	85	76	98.7	5.0	0.7
	75.6	T (M)	105	79	98.2	5.0	0.0
Still	85.1%	mean	88	75	98.4	4.4	0.3

[a] The surface area of the male subjects in the 3 and 4 men-group tests averaged 1.86 sq meters each. The surface area of the mixed subjects in the 5 and 6 group tests averaged 1.62 sq meters.

providing original data when the paper appears in the TRANSACTIONS is very much in order, and there will be presented with the written closure a large sample of data in their

original form for study and analysis by future investigators. Mention was made in the paper of subject impressions between morning and afternoon, and question was raised as to why these were not listed in detail. This could have been done. However, for use by the Society, it is felt that there is more interest in mass impressions than in the fluctuations of the group throughout periods of the day. Anyone using these data does not want to explore the idea as to whether the data are for afternoon or morning or evening. He is more interested in whether the data could be considered relatively representative for any of these periods. The authors think this is the case with the present data.

Dr. Bedford's plot (Fig. B) of the authors' data in comparison to some of the work which he did in Great Britain is very interesting. It is pleasing to note that the data checked as well as they did. The fact that the British people prefer lower temperatures than the authors' graphs indicate has been noted before, but whether this is a physiological or psychological condition remains to be seen. There is also a clothing difference which must be taken into consideration.

AUTHORS' CLOSURE, (WRITTEN): The authors have now had a chance to examine the written discussions and will comment on them. First it should be stated that a rather extensive selection of original data has been prepared and is presented here as Table A. This will give all who wish to study variants in the original data an opportunity to do so. The question has been raised as to whether in Figs. 5 and 7 of the paper enough data are available to extend the test curves to the comfort index, 4 value, or not. In answer to this, the authors certainly are of the opinion that the test data curves, if extended to the 4 value, should not go past this line, as the warm region alone should under no circumstances be used as the means of setting the comfort index. In fact, it seems most probable that the thermal sensation lines must flatten out with the 4-value occurring more over a range than at an intersection point. This cannot be predicted from tests in the warm region alone. Consequently, it is felt that any plot of the No. 4 line points on a comfort-type chart (dry and wet-bulb axes) could be misleading, and the fact that the authors' index 4 on such a chart is essentially perpendicular to the dry-bulb axis is inconclusive. However, subsequent testing at the Laboratory has indicated that the 4 line is essentially as steep as these tests would indicate it to be. It is asserted, therefore, that this is a fortuitous circumstance, and the authors would in no way expect to predict the 4-line slope from these data for which it represents an end of the testing range.

The authors do have confidence, however, in the trends of the 5, 6, and 7 points which are representative of the warm range, and would like to comment concerning Professor Nevins' comparison of some of his test results with theirs. Here it must be stressed that an absolute comparison of numerical values of subjective impression scales will hardly ever show the same results for diverse groups. Although it is relatively easy to find what represents a comfortable condition with any group, each distinct group will have its own interpretation of what is meant by slightly warm, by warm, or by hot; and, consequently, there will be variations in terms of these indexes. The basic relationship between the subjective impression and an objective unit is an arbitrary one.

During the progress of the tests, the authors made a number of statistical-analysis studies of the data in which correlation coefficients and regression lines were drawn. The correlation coefficients were sufficiently good to indicate that statistical reproducibility held, and in the one physiological factor of sweat rate (more accurately called evaporation rate), the authors' data were also in very close agreement with the work of prior experimenters. Consequently, it was decided to plot the averaged data exactly as they appeared and to include all of the tests for which there was certainty that the conditions in the room were maintained as indicated.

Concerning Mr. Rosen's comment, air at temperatures in the range of 98 F to 103 F does increase convective heat gain; but this effect, it is felt, is more than offset by the improved evaporative cooling effectiveness with air motion.

In reply to Mr. Flack's query, it is planned in the future to study variations in mean radiant temperature and response patterns to walls under asymmetrical conditions.

1656

No. 1656

SUPPLY OUTLET LOCATIONS FOR BASEMENT HEATING†

By J. R. WRIGHT* AND D. R. BAHNFLETH**, URBANA, ILL.

THE CEILING or high sidewall type of supply outlet has been commonly used in warm-air heating systems to heat basement rooms. Large room-air temperature variations and cool floors have resulted. This type of basement heat addition provides warm floors in first-story rooms[1, 2] but does not produce satisfactory comfort for extended occupancy in the basement space. The increasing use of basement rooms for recreational and living purposes suggested the need for an investigation to determine desirable supply outlet locations for heating these rooms. In addition, a related problem has been found in multi-level houses having rooms with concrete slab floors at or below grade. Large floor-to-ceiling temperature variations and cool floors occur in the lower level rooms of this type home when warm air is supplied to the rooms through overhead supply outlets. A warm-air perimeter system employing ducts embedded in the concrete slab floor would elimnate the unsatisfactory comfort conditions experienced. An embedded perimeter system may be used in new construction, but installation of embedded ducts is not economically feasible in existing structures. It was felt that a part of the results of this investigation would be applicable to multi-level type structures.

This investigation was conducted in Warm Air Heating Research Residence No. 2 for the purpose of satisfying the following objectives:

1. To determine the effect of supply outlet location in a basement room on:

(*a*) air temperature variation; (*b*) floor-surface temperatures; (*c*) room-air velocities; (*d*) fuel consumption.

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† This investigation was part of the cooperative project jointly sponsored by the Engineering Experiment Station of the University of Illinois and the National Warm Air Heating and Air Conditioning Association.

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** Assistant Professor of Mechanical Engineering, University of Illinois. Associate Member of ASHAE.

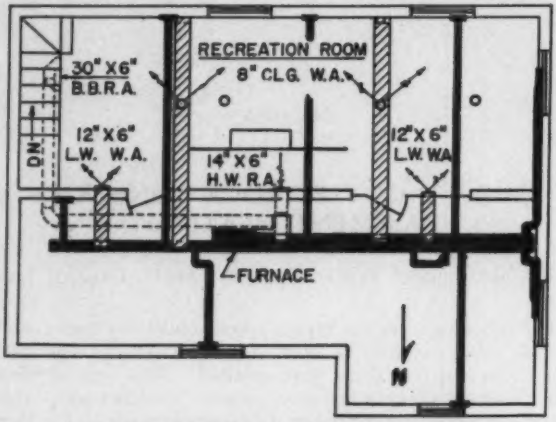
[1] Exponent numerals refer to References.

Presented at the 65th Annual meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, January 1959.

2. To determine the cyclical variation of basement-air temperature in a system in which the thermostat was located on the first story.

DESCRIPTION OF RESIDENCE

The residence, which has been described[1] in detail was a one-story frame structure with full basement. Walls were insulated with 3⅝-in. mineral wool blanket-type insulation with vapor barrier attached. Ceiling was insulated with 5 in. of



R.A. shows return air openings; W.A. shows warm air openings; B.B. shows baseboard openings; H.W. shows high sidewall openings; L.W. shows low sidewall openings; circles show locations of thermocouple standards.

FIG. 1. . . . BASEMENT PLAN AND DUCT LAYOUT OF FIRST-STORY SYSTEM (BLACK) AND BASEMENT BRANCH DUCTS (CROSS-HATCHED)

mineral wool. The calculated coefficient of heat transmission, U, for the walls and ceiling was 0.07 Btu per (hr) (sq ft) (F deg). Windows on the first story, with the exception of the double-glazed fixed window in the living room, were of the horizontal sliding type with storm sash attached to the window sash. First-story doors were of conventional wood and glass construction and equipped with storm doors. Windows were weather-stripped.

A room was made in the basement by erecting partitions in the openings between the south and north sections of the basement. The room is shown on the basement plan in Fig. 1. The partition was of 2 x 4-in. stud construction with a finish gypsum board panel on the test room side only. The test room in the south half of the basement had a floor area of 507 sq ft (39 x 13 ft) with glass areas of 33 and 16.5 sq ft on the south and west exposures, respectively. Basement windows were of the horizontal sliding type and were larger (56 in. wide x 41 in. high) than conventional basement windows. Two such windows were located in the south wall and one in the west wall. Basement windows were single glazed. There was no storm door on

the grade level entrance, which opened into the basement stairwell. The door and windows were weather-stripped.

The basement wall construction is shown in Fig. 2. The above grade portion of the wall consisted of approximately equal parts of 10-in. lightweight aggregate concrete block plastered outside and frame insulated with 3⅝-in. mineral wool insulation. The below grade portion of the walls consisted of 10-in. lightweight aggregate concrete blocks plastered outside. At window sections the surface of

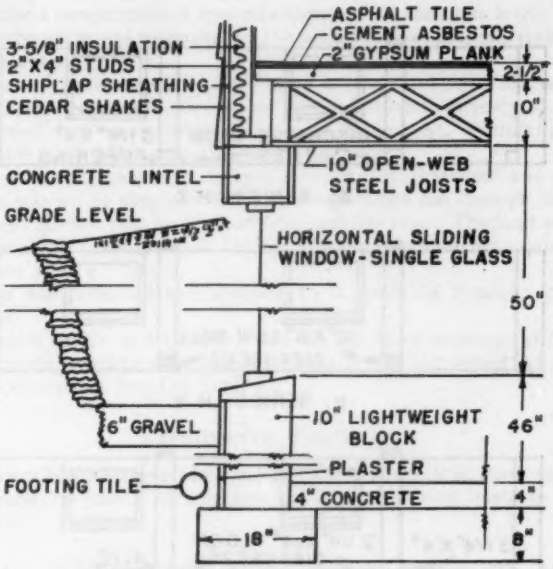


FIG. 2. . . .BASEMENT CONSTRUCTION

the concrete floor was approximately 3 ft below the surface of the gravel placed at the bottom of the light wells. At other sections the floor surface was approximately 7 ft 6 in. below grade. The east basement wall, with the exception of the frame portion and one course of block above grade, was protected by an attached garage and a concrete driveway.

The design heat loss of the basement and the stairway, calculated in accordance[3] with Manual 3, was 15,515 Btu per hr for design temperatures of −10 F outdoors and 70 F indoors. This amounts to 29.9 Btu per (hr) (sq ft) of floor area, which is approximately equal to the average heat loss of the lower level rooms investigated in field studies. The design heat loss of the first story was 31,047 Btu per hr, for a total design heat loss of 45,562 Btu per hr.

One major cause of temperature unbalance in multi-level houses is the interchange of air between floors through open stairwells. During heating, cool air from the

upper levels collects in the lowest level, while warm air from the lower level moves to upper levels. To allow this interchange of air in Research Residence No. 2, the door between the kitchen and the basement stairway was removed throughout the investigation.

DUCT SYSTEM AND BASEMENT SUPPLY OUTLET LOCATIONS

The small-pipe perimeter system which had been used during recent year-around air-conditioning investigations was used to heat the first-story rooms. The system,

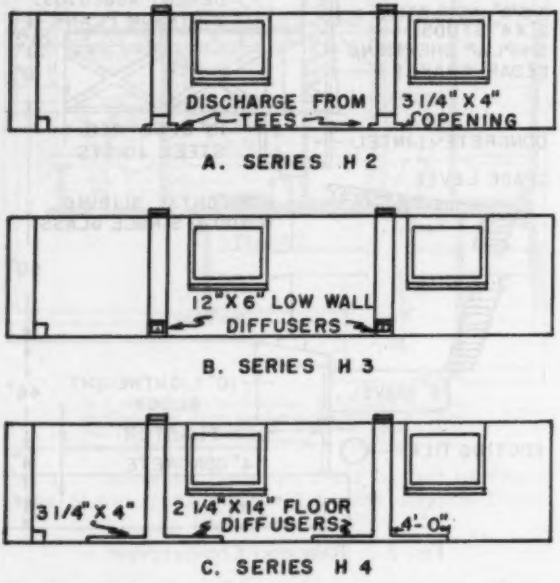

DISCHARGE FROM TEES
3 1/4" X 4" OPENING
A. SERIES H 2
12" X 6" LOW WALL DIFFUSERS
B. SERIES H 3
3 1/4" X 4"
2 1/4" X 14" FLOOR DIFFUSERS
4'-0"
C. SERIES H 4

FIG. 3. . . .PERIMETER WARM AIR SUPPLY OUTLET ARRANGEMENTS AT EXPOSED SOUTH WALL

shown in black in Fig. 1, was of the extended-plenum type having an 8 x 8-in. trunk and a 14 x 8-in. trunk extending east and west, respectively, from the furnace bonnet. The 9 branch ducts were of 4-in. diameter, and the supply outlets were 2¼ x 14-in. floor diffusers. The central return-air grille was located in the entrance hall on the first story.

The duct system in the basement, which is cross-hatched in Fig. 1, was designed to provide for the several supply outlet locations shown in Figs. 1 and 3. The locations include 2 ceiling outlets (see Fig. 1), free discharge from 2 ducts with tees to direct flow along the outside wall (Fig. 3A), two 12 x 6-in. low sidewall diffusers on the outside wall (Fig. 3B), four 2¼ x 14-in. floor diffusers on extended

ducts at the outside wall (Fig. 3C), and two 12 x 6-in. low sidewall registers on the inside wall (Fig. 1).

Two basement return locations, shown in Fig. 1, were used during the investigation. Only one of the basement returns, together with the first-story return, was used in any one series. The return not in use was sealed at the connection to the central return duct.

EXPERIMENTAL CONDITIONS

The heated space consisted of the first-story rooms and the room in the south half of the basement. The thermostat, which was located in the first-story was set to maintain a temperature of approximately 73 F at the 30-in. level. The basement-air temperature was controlled by the same thermostat as the system did not include zone control.

The fuel-input rate was set at 58,200 Btu per hr in all studies. This corresponded to the calculated heat loss divided by an assumed bonnet efficiency of 80 percent and an assumed duct transmission efficiency of 100 percent. Natural gas having a heating value of 1000 Btu per cu ft was the fuel used.

The system was adjusted in accordance with the CAC principle[4], and the system air-flow rate was set to obtain a 100 F air temperature rise through the furnace. The calculated air-flow rate for this condition was 430 cfm. The limit control was set to cut out at 200 F and cut in at 175 F. The fan switch was set to cut on at 100 F and cut out at 80 F.

The house was furnished and occupied by a family of 2 adults during the investigation.

The basement supply outlet location and the basement return-air grille location were the controlled variables in the investigation. The test series and the supply and return locations are listed in Table 1.

EXPERIMENTAL PROCEDURE

All significant temperatures, electrical and gas consumption, and weather variables were recorded daily. In addition, special studies were made to determine

TABLE 1. . . .SUPPLY OUTLETS AND RETURN-AIR COMBINATIONS STUDIED

TEST SERIES	BASEMENT SUPPLY OUTLETS			BASEMENT RETURN-AIR GRILLE
	No.	TYPE	REMARKS	
H 1	2	Ceiling Diffusers		Baseboard[a]
H 2	2	Open Tees	Free discharge along outside wall	Highwall[b]
H 3A	2	Low Sidewall Diffusers	Air directed upward along outside wall	Highwall
H 3B	2	Low Sidewall Diffusers	Air directed onto floor	Highwall
H 4A	4	Floor Diffusers	On extended ducts at outside wall	Highwall
H 4B	4	Floor Diffusers	On extended ducts at outside wall	Baseboard
H 5	2	Low Sidewall Registers	On inside wall at floor level	Highwall

[a] Baseboard return was located at base of stairway to the first story.
[b] Highwall return was located near the center of the inside basement wall at ceiling level.

basement-air temperature and velocity variation in the floor and sitting level planes, cyclical variations in basement and first-story air temperatures under various load conditions, supply-air characteristics, and branch air-flow rates.

Comfort depends upon many variables, but in this paper the systems studied have been evaluated on the basis of room-air and floor-surface temperatures and room-air velocities. The subjective factors of odor, dust content and noise could not be evaluated.

At the beginning of each series, the branch dampers were adjusted to obtain satisfactory temperature balance between the basement and first story, and between rooms. These adjustments were made during average winter weather and under night-time operating conditions to avoid the influence of solar radiation. All temperature data were collected at 6:30 a. m. when the effects of sun and occupancy were negligible, and all temperatures reported, unless otherwise noted, were based on these readings.

RESULTS

The several types of supply outlets used may be divided into 3 groups according to the way in which conditioned air was discharged into the space. The 3 groups and the types of outlets included in each group are: horizontal discharge at ceiling level, ceiling diffusers; horizontal discharge at floor level, inverted low sidewall diffusers, open tees and low inside wall registers; and vertical discharge at floor level, floor diffusers on extended ducts and low sidewall diffusers. The performance of the ceiling diffusers, low inside wall registers and floor diffusers was representative of the outlets in each of the 3 groups and, in general, only the studies with these outlets are discussed.

Supply and Room Air Velocities: The *supply air characteristics* for floor diffusers on extended ducts (Series H 4) and low inside wall registers (Series H 5) are shown in Fig. 4. Isovels are indicated for velocities of 100 and 200 fpm. Supply air characteristics were obtained by measuring velocities at a number of elevations and distances from the centerline of the west basement supply outlet. The 100- and 200-fpm jet isovels were then graphed. The burner was permitted to operate during the time that the velocities were measured, and the average supply-air temperatures for the series shown were 95 F and 93 F in Series H 4 and H 5, respectively.

The ceiling diffusers distributed air equally in all directions and the primary air was parallel to the ceiling. Velocities beneath the outlet were not of sufficient magnitude to be noticeable. The air in the living zone was nearly stagnant, which is in agreement with the results[5,6] of a laboratory study of air distribution.

The floor diffusers installed at the end of the extended ducts (Series H 4) had vanes which were deflected progressively from 0 deg at the center to about 42 deg at the ends. The primary air was not spread as would be expected with this type of outlet because the air was delivered to the outlet from one end. The deflection of the single jet suggests that most of the primary air passed through that half of the vanes having the same deflection as the jet. The 100-fpm isovel was approximately 5 ft above the floor at its highest point.

The vanes of the conventional low sidewall registers used on the inside wall (Series H 5) were deflected approximately 22 deg on each side of center, and a 2 jet outline was obtained at the 1-in. level. The throw of the west half of the jet was less than that of the east half because of the influence of a nearby cabinet. The 100-fpm isovel of the east jet extended 6 ft into the room at the 1-in. level, and at its

widest point the 100-fpm isovel was about 7 ft wide. The maximum velocity measured at the 6-in. level was 80 fpm, and the average at the same level and immediately above the primary air was 50 fpm.

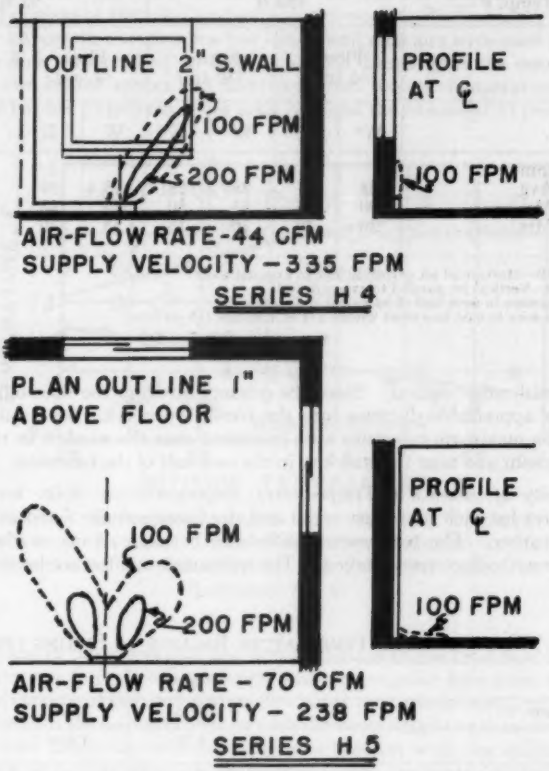


FIG. 4. . . .SUPPLY AIR CHARACTERISTICS FOR FLOOR DIFFUSERS ON EXTENDED DUCTS (H 4) AND LOW INSIDE WALL REGISTERS (H 5)

Basement-air velocities at the 4-in. and 30-in. level measured in the west half of the basement and in the east end of the basement within 4 ft of the stairway during 2 test series are listed in Table 2. The maximum velocities at the 4-in. and 30-in. levels with horizontal discharge at floor level (Series H 3B) were measured directly in front of the outlet. The average velocity at the 4-in. and 30-in. levels was not excessive. The basement-air velocities measured with the floor diffusers located on the top of extended ducts were generally smaller in magnitude than those

TABLE 2. . .BASEMENT-AIR VELOCITIES AT SITTING AND FLOOR LEVELS

SUPPLY OUTLET	HORIZONTAL DISCHARGE AT FLOOR[a]				VERTICAL DISCHARGE AT FLOOR[b]			
Supply-air Temp. F	103.0				95.0			
Level	Floor (4 in.)		Sitting (30 in.)		Floor (4 in.)		Sitting (30 in.)	
Stations	W[c]	E[d]	W	E	W	E	W	E
Velocities, fpm								
Avg	38	31	28	28	25	29	26	27
Max	90	45	45	50	45	45	33	60
Min	20	20	20	15	18	20	20	20

[a] Series H 3B—Horizontal jet perpendicular to exposed wall.
[b] Series H 4—Vertical jet parallel to exposed wall.
[c] W stations were in west half of basement (35 stations).
[d] E stations were in east basement within 4 ft of staircase (18 stations).

measured with other outlets. Since the primary air from the floor outlets did not extend any appreciable distance into the room, lower velocities would be anticipated. The maximum velocities were measured near the window in the west half of the basement and near the stairway in the east half of the basement.

Uniformity of Room-Air Temperatures: Representative 30-in. level room-air temperatures for each first-story room and the basement are listed in Table 3 for average weather. The temperature difference between rooms on the first story was similar with all systems studied. The temperature difference between the first-

TABLE 3. . . .ROOM-TO-ROOM TEMPERATURE BALANCE AT SITTING (30-IN.) LEVEL

SERIES	H 1	H 4	H 5
OUTDOOR TEMP, F	23.5	24.8	28.5
WIND, MPH	8 W	CALM	5 W
ROOM-AIR TEMPERATURES AT SITTING LEVEL, F			
Living Room East	70.6	72.2	73.1
West	70.4	72.2	73.3
South Bedroom	70.4	72.6	72.6
Bath	72.0	73.5	73.1
North Bedroom	72.0	73.1	72.6
Dining Room	71.6	73.1	73.5
Kitchen	71.8	72.8	72.8
Southwest Basement	67.1	72.8	73.5
Southeast Basement	68.0	74.0	73.8
Maximum Difference			
First Story	1.6	1.3	0.9
House	4.9	1.8	1.2

story room and the basement, based upon data collected at 6:30 a.m., was satisfactory except when the ceiling outlets were used in the basement. It was found with ceiling diffusers that little improvement in temperature balance could be made by increasing the air flow to the basement ducts. Stratification occurred in the basement, and warm air was observed moving up the stairway into the first story.

About 62 percent of the total air flow was delivered to the first story and 38 percent to the basement, except when low inside wall registers were used and 67 percent of the flow was required on the first story. Based upon the calculated heat loss of the two heated spaces and assuming equal register temperatures, the first story would require 67 percent of the total flow and the basement 33 percent. The

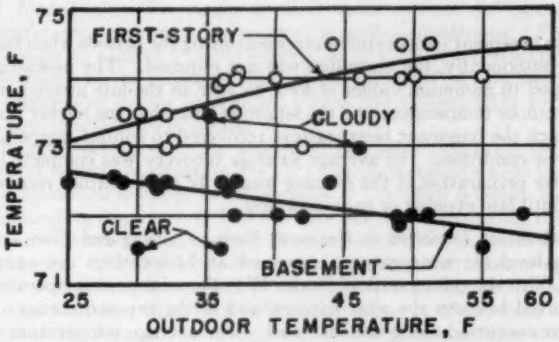


FIG. 5. . . . EFFECT OF AVERAGE DAILY OUTDOOR TEMPERATURE ON AVERAGE DAILY ROOM-AIR TEMPERATURE AT THERMOSTAT ON FIRST STORY AND AT 30-IN. LEVEL IN BASEMENT (H 4)

small differences in the theoretical and actual division of the total flow were attributed to variations in register-air temperatures and vagrant heat gains to the two spaces. It should be noted that the air-flow rates to the first story and basement were the same with the ceiling outlets as with the other supply outlets, even though the temperature difference between floors was greatest with the ceiling outlets. The difference in performance was attributed to the movement of the stratified basement air up the stairway to the first story.

Room-Air Temperature Variations During Cycling of Burner and Blower: Cyclical variations in room-air temperature during continuous blower operation and cyclical burner operation amounted to about 0.4 F deg on the first story and 0.7 F deg in the basement. During periods of high solar intensity in mild weather, the reduction of the first-story heat loss caused cycling of the blower, and the basement-air temperature dropped because the effect of sun on the basement heat loss was comparatively small. Average daily temperatures at the thermostat on the first story and at the 30-in. level in the basement have been plotted in Fig. 5 as a function of average daily outdoor temperature. These data were obtained with floor diffusers. The trends of the curves indicate that the extraneous heat gains (such as sun and cooking) were equal to or greater than the first-story heat loss during mild weather,

TABLE 4. . . . BASEMENT-AIR TEMPERATURES AT FLOOR (4-IN.) AND SITTING (30-IN.) LEVELS

SERIES	OUTDOOR TEMP, F	SUPPLY TEMP, F	FLOOR LEVEL TEMPS, F			SITTING LEVEL TEMPS, F		
			AVG[a]	MAX	MIN	AVG[a]	MAX	MIN
H 1	9.3	110.0	66.6	67.8	64.3	67.8	68.4	67.3
H 4	17.0	117.0	71.9	72.8	70.0	73.7	74.7	72.8
H 5	33.2	106.0	72.7	77.3	70.4	73.7	74.9	72.8

[a] Averages are for 73 stations.

and that the basement temperature decreased during the periods when burner operation and, consequently, fan operation was not required. The basement temperatures dropped to minimum values of 69 F to 70 F in the late afternoon depending upon the outdoor temperature and the length of time that the burner was off. The time at which the basement temperatures recovered to control temperature varied with outdoor conditions. In average weather recovery was complete in the early evening after preparation of the evening meal. In mild weather recovery was not complete until late evening or early morning.

Air Temperature Variation in Basement Room at Sitting and Floor Levels: Floor- and sitting-level air temperatures measured at 73 locations are summarized in Table 4. With the ceiling outlets (Series H 1) the minimum temperatures at both levels occurred beneath the west window and in the exposed corner. Maximum temperatures occurred along the east wall. The average temperature was low for the reasons discussed in the section on temperature balance.

With supply outlets low along the south wall (Series H 4) minimum temperatures were found beneath the west window, and maximum temperatures were found near the east supply outlet. The small difference between maximum and minimum temperatures with the floor diffusers at the end of extended ducts was due to the fact that the primary air did not extend into the room.

Minimum temperatures were found beneath the south and west windows with low sidewall registers at the inside wall (Series H 5). Maximum temperatures occurred near the supply outlets. Comparison of maximum and minimum temperatures at the 4-in. and 30-in. levels with the low inside wall outlets indicates that the supply air had the greatest effect on the temperature distribution at the floor level.

Room-Air Temperature Differentials and Temperature Gradients: The living zone (4-in. to 60-in. level) temperature differentials were satisfactory with all systems studied, being less than 5 F deg at design conditions. Temperature differentials are listed for the 3 outlet groups in Table 5 for 2 indoor-outdoor temperature differences. The smallest living zone differentials were produced by outlets having horizontal discharge at floor level. The larger differentials obtained with the floor diffusers were attributed to the lack of spread of the primary air. As a result, more stratification occurred with these outlets, and the exposed wall was not blanketed with warm air. While the living zone temperature differentials were small with ceiling diffusers, the temperatures were low.

Representative temperature gradients are given in Fig. 6 for the 3 outlet groups at outdoor temperatures in the range from 31.8 F to 33.1 F. The gradients are

those measured in the southwest basement. When the supply air was discharged horizontally at the ceiling, stratification occurred and the temperatures above the 60-in. level were high. The movement of this warm air from the basement ceiling into the first story was noticeable, especially during cold weather. The gradient in the living zone was small, but the air temperatures and the floor-surface temperatures were low. The most satisfactory gradient was produced by those outlets discharging the air horizontally along the floor; however, the gradients with vertical discharge supply outlets were not appreciably larger. The low floor-level air temperatures indicate that heat addition to the concrete slab floor and more complete blanketing of the exposed walls with warm air would be required to improve the gradients.

Basement Floor-Surface Temperatures: With each system the basement floor-surface temperatures were nearly independent of outdoor conditions. For example, the daily averages at each of 5 points selected to give a good representation of the different conditions on the basement floor varied less than 3 F deg over the range of weather experienced during each series. Also, the temperatures were approximately constant over any 24-hr period.

Floor-surface temperatures in the southwest basement (approximately one-half of the basement room) are shown in Fig. 7. The temperatures were plotted from 6:30 a.m. temperature readings at 23 stations. The days selected had outdoor temperatures ranging from 24.8 to 28.5 F. The effect of introducing the warm air at floor level is evidenced by comparing the temperatures of Series H 1 with those for the other series. The floor-surface temperature near the center of the room in this series was 2 to 5 F deg lower than in the other series, and the floor area shown was at a temperature less than 69 F. With the other systems, from one-half to three-quarters of the floor surface shown was at a temperature above 70 F. The lowest floor-surface temperatures were found in the exposed corner and along the west wall. Although the outlets with horizontal discharge produced the highest floor-surface temperatures in the center, the cool floor area along the west wall was larger than that obtained with the floor diffusers on the extended ducts.

FUEL CONSUMPTION

Fuel consumption was less when the ceiling outlets were used (Series H 1) because of the lower temperatures maintained in the basement with these outlets. The highest fuel consumption was observed with the low inside wall registers (Series

TABLE 5. . . .AVERAGE BASEMENT-AIR TEMPERATURE DIFFERENTIALS

INDOOR-OUTDOOR TEMP DIFF, F	35			70		
SERIES	H 1	H 4	H 5	H 1	H 4	H 5
	Temperature Differentials, F					
4 in. to 30 in. Level.....	1.4	2.7	1.7	2.4	2.8	2.6
4 in. to 60 in. Level.....	3.3	4.1	2.2	3.9	4.8	3.1
4 in. to 92 in. Level.....	8.6	5.3	2.6	15.2	8.0	3.8

H 5); however, since the data for this system were collected during mild weather, the data obtained by extrapolation are not conclusive. The increase in fuel consumption resulting from heating the basement may be obtained by comparison of the data for the Series H tests with that for a previous heating season when heat

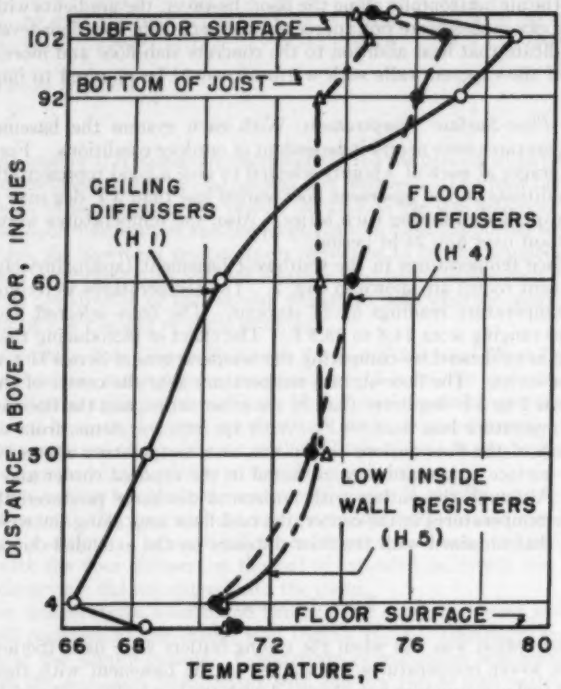


FIG. 6. . . .BASEMENT-AIR TEMPERATURE GRADIENTS IN SOUTHWEST BASEMENT. WITH OUTDOOR TEMPERATURES FROM 31 TO 33 F AT NIGHT; WITH CONTINUOUS BLOWER OPERATION; WIND VELOCITY OF 3 TO 5 MPH, SOUTHERLY

was not added directly to the basement. When the supply outlets were located to supply air at or near the basement floor level, the fuel consumption increased 17 percent over that for the unheated basement at about 0 F outdoors (indoor-outdoor temperature difference = 70 F deg). The increase in fuel consumption over that for the unheated basement amounted to about 8 percent at 0 F outdoors when the ceiling outlets were used.

There was some concern at the beginning of the investigation that placing the partition wall between the 2 halves of the basement would reduce the amount of

combustion air available to the furnace, thereby causing poor venting of the furnace with subsequent spillage at the draft hood. A thermocouple grid was placed in the draft hood and the temperature at the grid was recorded during the entire investigation to detect spillage. On the basis of the recorded temperatures and CO_2 content of the flue gas, it may be concluded that the addition of the wall had no measurable effect on the supply of combustion air to the furnace. Infiltration through the three windows in the room in which the furnace was installed and the movement

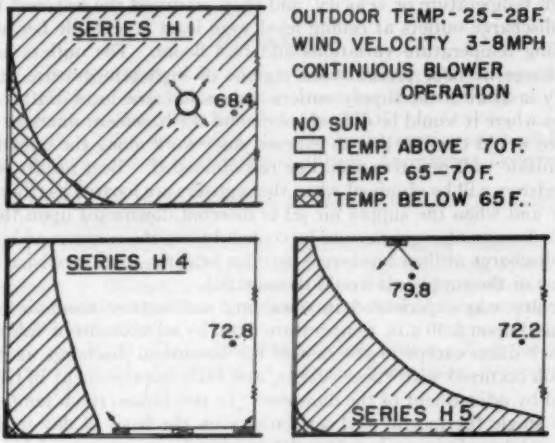


FIG. 7. . . .FLOOR-SURFACE TEMPERATURES IN SOUTHWEST SECTION OF BASEMENT

of air beneath the undercut doors separating the 2 basement spaces was apparently sufficient to satisfy the air requirements of the furnace.

BLOWER OPERATING CHARACTERISTICS

The pressure loss on the warm air side of the ceiling supply system was approximately 0.065 in. of water for the total flow rate of 430 cfm. Addition of the extended ducts and floor diffusers at the outside walls caused an increase of about 100 percent in pressure loss on the warm air side. The pressure loss on the warm air side with the low inside wall registers was 50 percent greater than that in the first series. The increased pressure loss with the inside wall outlets was a result of the increased air flow to the first story required to achieve satisfactory balance with this system. The static pressure loss on the return air side was nearly constant in all series. The total external pressure losses, which ranged from 0.100 to 0.175 in. of water, were less than design values. The increased pressure loss with the Series H 4 and H 5 systems was accompanied by an increase in blower power requirement.

The blower operated approximately 19 hr at an average indoor-outdoor tempera-

ture difference of 30 F deg. Continuous operation of the blower occurred at indoor-outdoor temperature differences greater than 50 F deg (23 F outdoor temperature).

DISCUSSION

Outlets with vertical discharge at the outside wall provided highest level of comfort in the basement room. Although the temperature differentials with these outlets were slightly greater than those with horizontal discharge at floor level, the vertical discharge outlets would be favored because they did not cause any localized areas of high temperature or velocity, and they produced the warmest floor. The horizontal discharge outlets at ceiling level were least acceptable because of large floor-to-ceiling temperature variations and cool floors. The outlets having horizontal discharge at floor level caused regions of high temperature and velocity immediately in front of the supply outlets and could cause local drafts. However, in situations where it would be difficult to extend the basement ducts to the outside wall or there would be objection to exposed duct work along the outside wall, the use of low inside wall registers would be recommended. Best results with low inside wall registers will be obtained when the registers are located as close as possible to the floor and when the supply air jet is directed downward upon the floor surface. If the basement room were to be cooled during the summer and outlets with horizontal discharge at floor level were used for heating, some provision for varying the direction of the supply air would be essential.

No difficulty was experienced in obtaining satisfactory room-air temperature balance, based upon 6:30 a.m. temperature data, by adjustment of volume dampers in the branch ducts except in the case of the horizontal discharge at ceiling level. Stratification occurred with these outlets, and little improvement in balance could be obtained by adjustment of the dampers. In this house, room temperature control with a single thermostat was satisfactory on the basis of day-to-day changes in the temperature balance between floors. No large amount of lag or overrun of basement-air temperatures occurred following large variations of outdoor temperature. The need for individual control of the basement temperature was most evident during periods of light load when the first-story heat loss was satisfied by solar heat gains and internal heat loads from cooking. During such periods the burner and blower did not operate and the basement-air temperature decreased to unacceptable levels.

The high room-air velocities, low room-air temperatures, and cool floor surface beneath the window in the west wall of the basement indicate the need for supply outlet under each window.

CONCLUSIONS

1. When heating basement rooms with warm air, the supply outlets should be located to deliver the air at floor level, preferably near the perimeter of the room to blanket the exposed walls and windows with warm air. When outside wall locations are not feasible, the heated air should be delivered from low inside wall locations.

2. Zone control may be desirable when heating a basement room, but the need for it will depend upon the use for which the room is intended and the thermal characteristics of the room. In the room investigated the need for zone control was evident only during mild sunny weather when the burner and, consequently, the blower did not operate.

3. The increase in fuel consumption caused by heating the basement room was less than would be anticipated from the increase in calculated heat loss of the heated space

because the basement was normally heated to some degree by vagrant heat losses from furnace casing, smokepipe and ducts. Outlets which delivered the air at floor level required a greater fuel consumption than those that delivered the air at ceiling level, but the space temperatures were also higher.

ACKNOWLEDGMENTS

It is expected that these results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is hereby made to the manufacturers who cooperated by furnishing materials and equipment.

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DISCUSSION

H. H. REICH, Pittsburgh, Penn.: I read this article when it first appeared in the JOURNAL. I was working on a job at the time and was getting worried because of being forced into using high sidewall supplies, which, at best, would not have been any better than the ceiling supplies that were used in the test. But, due to the construction of the building, I was able to use low outside returns, and quickly turned to the paper and did not find any tests for low outside returns, that is, baseboard returns.

Being forced into using high sidewall supplies, I felt that I was making the best of the situation by using the low baseboard outside return, and was wondering in all fairness to the disadvantages of the high supplies, why, to get the best results, low outside returns were not tested. In other words, the low inside returns which were tested, I don't think, would have given as good results and made the best of a bad situation as low outside returns. If there has not been some previous investigation on that score, it is suggested that it be made to see if this situation of the high sidewall supplies could be improved.

H. B. NOTTAGE, Encino, Calif.: Referring to Fig. 6, giving temperature profiles above the basement floor, it is significant to observe that the floor surface temperature is always above the air temperature in the vicinity of the floor. This means that the floor of the basement acts as a source of heat flow to the air by convection.

The house sub-floor temperature is seen to be always above the basement floor surface temperature, so that heat radiates downward to the basement floor. Hence, for practical comfort and economy it is wondered how this effect could be most advantageously combined with air distribution. What might the authors propose?

Another practical question is that of moisture. When air sweeps concrete floors and walls below grade, these surfaces, in the heating season, and unless perfectly sealed, act as moisture sources. This moisture addition can be quite detrimental. What thoughts can be advanced on moisture control in relation to air distribution? In the cooling season, what about moisture condensation?

The entire matter of interior finish for basement walls and floors is significant in relation to air distribution, thermal balance, and comfort. What does the continuing program envision in this connection?

K. R. SOLVASON, Ottawa, Ontario, Canada: I would like to ask the authors at what part of the winter heating season these tests were run, that is, near the beginning or toward the end. One would assume that the basement heat loss becomes a greater proportion of the total house loss towards the end of the winter heating season than at the beginning. This could presumably influence the basement temperatures resulting with control by an upstairs thermostat.

AUTHORS' CLOSURE (Mr. Wright): I should first like to answer the question posed by Mr. Reich regarding high sidewall supply outlets and outside lowwall return intakes. A system of this type was not tested. There is published information available from which I would like to extract two statements: "Much more was done to alter the air distribution by changing the supply outlet than by changing the return intake location." Also, "if an outlet produced uniform temperature distribution with little stratification, the location of the return intake had only a limited effect on the distribution." Rarely, if ever, can one correct a poor comfort condition by location of return intakes. This information is from an investigation conducted at the University of Illinois and the results are published in 2 University of Illinois Engineering Experiment Station Bulletins, Nos. 435 and 442 (see References 5 and 6 of this paper). All types of supply outlets and return intake locations were investigated.

Mr. Nottage's suggestion that the basement floors might be heated by radiation from the floor above certainly merits consideration. Maximum comfort conditions would be achieved by adding heat at the base of the outside walls to counteract the cool air currents.

As far as moisture in the basement is concerned, no difficulty was experienced. The exposed walls had been plastered on the outside and had a bituminous sealer applied to them. The slab was over gravel but did not have a vapor barrier. There was no noticeable movement of moisture into the basement from either the walls or the floor. Relative humidity records were maintained and though they are not included in this paper there was not a significant difference in the relative humidity between the basement and the first story.

Mr. Solvason questioned the part of the heating season in which these tests were conducted. They were in chronological order as listed in the paper from Series H1 through H5, beginning about the first of December and ending in March. Therefore, the perimeter diffusers with vertical discharge low at outside walls produced the best comfort conditions during the most severe part of the heating season; that is, in February and March when the ground temperature was near its minimum. Floor surface temperatures and wall surface temperatures indicated a minor effect of time of season on the conditions in the basement.

1657

No. 1657

PULSATIONS IN SINGLE-PORT GAS-FIRED RESIDENTIAL HEATING EQUIPMENT

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PULSATIONS in heating equipment have plagued manufacturers and users for many years. The suppression or elimination of these pulsations has led to the evolution of a large number of suppression techniques. Unfortunately, no one technique seems to be adaptable to all pulsation problems While users were becoming more noise conscious, the entire problem has been aggravated by the trend toward placing smaller, tighter heating units nearer the living areas of the residence. Need has therefore arisen to find the basic cause of the pulsations and thereby obtain better ways of *curing* the oscillations. To implement this search for the basic causes of pulsation, a research program was initiated at Battelle in May, 1954, to (1) determine the basic mechanism of oscillation in both gas- and oil-fired heating units, and (2) determine means of suppressing or eliminating these oscillations or pulsations. During the 4 years of this program, studies have been made of oscillations in gas-fired multiple-port, gas-fired single-port, and oil-fired heating units. Numerous investigations of all 3 types of units have been made both in the field and in the Battelle laboratories. During this time, many different variables or parameters have been studied to determine what effect each had on the oscillations or pulsations of the units. From these studies, the basic mechanism for the oscillations in each of these units has been postulated.

This is the fourth in a series of papers presenting the findings of this project[1, 2, 3]. The purpose of this paper is to outline the basic mechanism of oscillation for single-port gas-fired heating units. Included are the findings leading up to the postulation

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[1] Exponent numerals refer to References.
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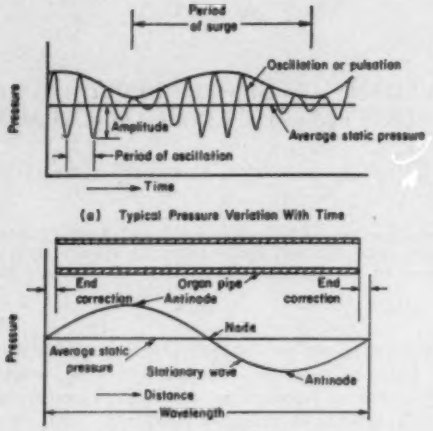
of the mechanism, a description of the mechanism from a macroscopic viewpoint, and additional data which appear to confirm this mechanism. As a preface to a succeeding paper, the application of this mechanism to the suppression of single-port gas flame oscillations is discussed briefly.

TERMINOLOGY

To aid in discussing the theoretical considerations and the results of the tests, a number of acoustical terms will be defined.



(a) Typical Pressure Variation With Time

(b) Typical Pressure Distribution in Space for a Stationary-Wave System (Organ Pipe)

FIG. 1. . . .PRESSURE VARIATION IN AN OSCILLATING GASEOUS SYSTEM

Fig. 1 shows the sinusoidal variations of pressure with time and space, and those terms which may be clarified by illustration are shown on the figure.

1. *Oscillation:* periodic variation, with time, of pressure with respect to the ambient pressure. During oscillation, the pressure is alternately greater than and less than the ambient pressure.

2. *Pulsation:* here considered to be the observable effects of the pressure oscillations. That is, pulsations can be seen, felt, or heard by an observer, or measured as changes in pressure of the surrounding atmosphere or as forces exerted on surrounding surfaces. Since there are certain physiological implications connected with the term pulsation, the more precise term oscillation is used in the subsequent discussion.

3. *Surge* (also called burst, thrust, thump): the amplitude of oscillation usually varies with time; the *surges* of high amplitude often occur about one to five times a second. The ear hears these surges as variations in noise or pulsation amplitude. The surges may be accompanied by a change in overall pressure level in the heating unit. This results in blasts of air pressure which can rattle loose parts and even loosen objects in the furnace room.

4. *Cycle:* complete sequence of values of pressure; at the end of a cycle, the pressure, the rate of change of pressure, and the direction of change of pressure have the same values that they did at the start.

5. *Period:* time interval corresponding to one cycle.

6. *Frequency:* number of periods occurring in one second or number of cycles per second (cps).

7. *Amplitude:* absolute value of the maximum difference between the instantaneous static pressure and the ambient pressure in a given period. The variations in the instantaneous static pressure can be with respect to time or space and the value of the amplitude need not be the same for both.

8. *Phase:* fractional part of a period through which the independent variable, usually time, has advanced, measured from an arbitrary origin. The origin is usually chosen so that the fraction is less than one.

9. *Wavelength:* distance between 2 wave fronts in which the pressures have a difference in phase of one complete period. During one period, the wave has advanced a distance equal to the product of the velocity of sound by the period.

10. *Mode Number:* the mode number is equal to the number of half wavelengths along a given dimension for a given system. The fundamental mode of a simple system represents the situation where the mode number for the system is equal to one for each dimension under consideration.

11. *Standing Wave:* oscillations having a fixed distribution of amplitudes in space, and characterized by the existence of nodes and antinodes that are fixed in space. In the fundamental mode of a heating unit, the acoustic pressures at all points are in phase.

12. *Nodes:* locations in a standing wave where the instantaneous pressure is equal to the ambient pressure during a given period.

13. *Antinodes:* locations in a standing wave where the instantaneous static pressure is equal to the amplitude during a given period.

14. *Natural Frequency:* the frequency at which a system will tend to oscillate when disturbed. The natural acoustic frequency of an enclosed space is a function of the configuration and volume of the space.

15. *Resonance:* for this discussion, resonance can be considered to be a condition in a system where the frequency of a periodic driving force on the system is equal to the natural frequency of the system. The effect of resonance is to cause oscillations of large magnitude in the system and to allow coupling for maximum energy transfer to take place between the system and the driving force.

16. *Resonant Frequency:* the frequency of the periodic driving force which causes a system to resonate.

17. *Normal Frequency:* for this discussion, normal frequency denotes the frequency of vortex shedding from the lip of the burner tube when the burner is external to the furnace.

18. *Helmholtz Frequency:* a resonant frequency of an enclosure made up of one or more nearly closed volumes, and one or more openings or connections of small cross-sectional area. The magnitude of this frequency is dependent upon the shape of the system and the mode of oscillation.

19. *Helmholtz-type Oscillation:* an oscillation occurring within an enclosure at or near the Helmholtz frequency.

20. *Damping:* the attentuation of the amplitude of oscillation which results from frictional and acoustical radiation effects in the system.

21. *Coupling:* an interaction between systems. In this paper, the interaction is between the vortex shedding and the pressure oscillation in the heating unit. When there is little interaction, the coupling is said to be low or *loose*; when there is large interaction, the coupling is said to be high or *tight*.

22. *Percent Primary Theoretical Air:* the weight ratio, multiplied by 100 of the actual primary air flow to the minimum air flow needed for theoretical complete combustion. For all of the investigations of single-port burners during this project, natural gas was used as the fuel. Upon the advice of the Pulsation Research Steering Committee, the stoichiometric air-fuel ratio for natural gas was taken to be 9.8 to 1.

INITIAL INVESTIGATIONS

As a starting phase, trips were made into the field and to laboratories of various manufacturers of residential gas-heating units. By investigating a number of different units, it was determined that the frequency range of the oscillations occurring in heating units fired with single-port gas burners was between 40 and 80 cps,[1, 2] and were of the standing-wave type. In addition to these field studies, a conventional type of single-port gas unit with a drum-type heat exchanger was investigated more extensively in the laboratory. These additional studies confirmed the field test results that the oscillations were of the standing-wave type. The experience of manufacturers and members of the Pulsation Research Steering Committee, plus the results of these studies, indicated that there were a number of variables which could have some effect on the amplitude of oscillation produced by the heating unit.

Fig. 2 shows the experimental single-port gas furnace constructed to permit the more complete investigation of the effect of various parameter changes on oscillations. This furnace was free from internal complexities. All of the factors investigated and the results obtained are presented in detail in Appendices B and C. Of the variables investigated, only the rate of primary air flow had any large effect on the amplitude of oscillation. Other variables such as the secondary air opening, which controls the rate of secondary air flow, were found to have a small effect on the amplitude of oscillation. This effect was later attributed to changes in the acoustic system whereby both the resonant frequency of the furnace and the position of the pressure antinode inside the furnace were controlled partially by the secondary-air opening. If changing the size of this opening caused the pressure antinode to approach the flame, then an increase in the amplitude of oscillation would be noticed. However, this change in amplitude was not nearly as large as that experienced with changes in primary-air flow.

A change in the length of the burner tube was also found to affect the amplitude of oscillation. This effect will be discussed subsequently in greater detail.

Other parameters such as spreader diameter, spreader height, and position of the burner in the furnace were found to have little or no effect on the amplitude of oscillations. Since changes in primary air-flow rate caused the only large change in the amplitude of oscillations, it became apparent that the mechanism of oscillation was concerned with events which occurred at the burner lip or in the burner tube.

At the beginning of these studies, high-speed movies were taken of the flame with the burner inside the furnace. Analysis of these photographs indicated that there were periodic disturbances in the flame which had the appearances of vortices being shed from the lip of the burner tube. Photographic sequences of this phenomenon were shown in the first report of this series[1]. The indication that vortex shedding from the burner might be present was considered an important step in finding the basic mechanism of oscillation. Therefore, additional high-speed movies were taken. These movies were taken of the flame with the burner external to the furnace and in a large room, in order to eliminate the effect of the furnace as a variable.

Fig. 3 shows a typical photographic sequence of the vortex shedding under the condition of the burner in a large room. These vortices indicated by the circles on

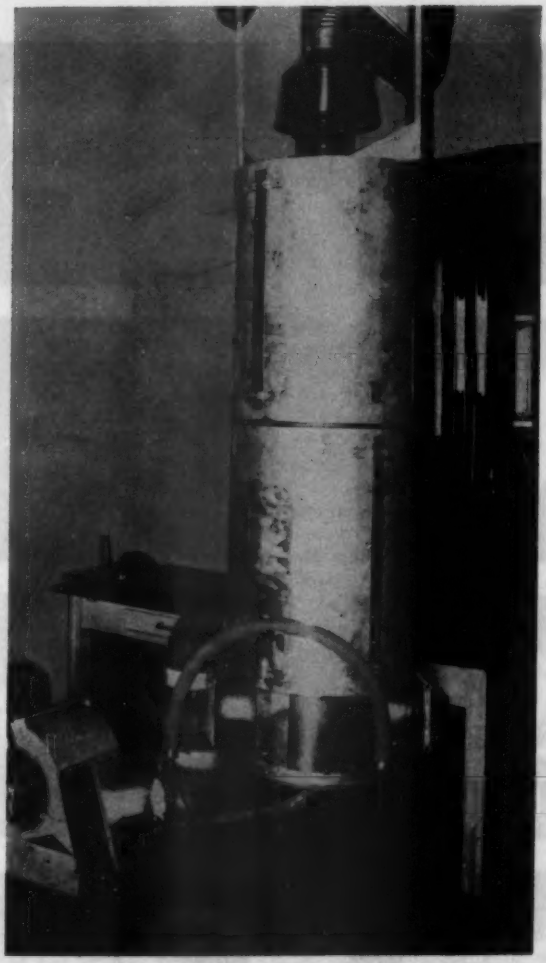


FIG. 2. . . .EXPERIMENTAL GAS-FIRED FURNACE FOR OSCILLATION STUDIES

photographs 2 through 8, appear like smoke rings passing up through the flame and causing undulations along the envelope of the flame. Different air and fuel-gas flow rates were used to determine to what degree the frequency of the vortex shedding was dependent on the velocity of the combustible gas at the burner port and on the mixture ratio. No effect of mixture ratio was found.

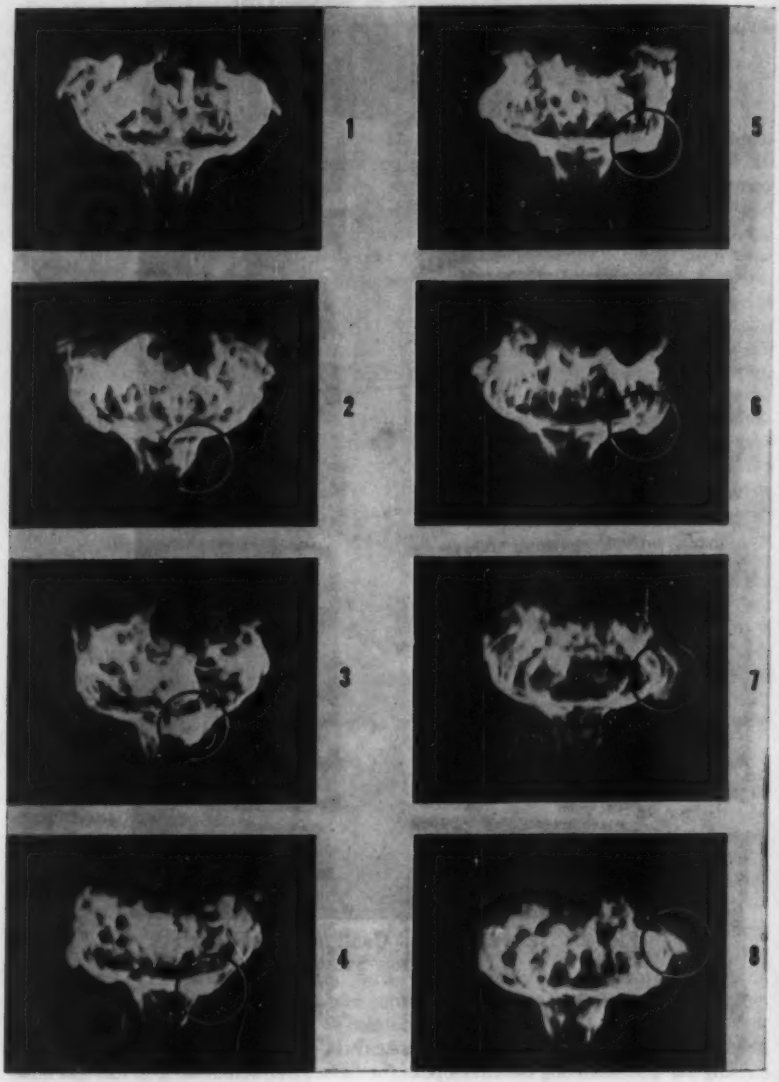


FIG. 3. . . .SEQUENCE OF HIGH-SPEED PHOTOGRAPHS SHOWING THE SHEDDING OF VORTICES FROM A SINGLE-PORT BURNER EXTERNAL TO THE FURNACE (58 PERCENT PRIMARY THEORETICAL AIR)

THEORETICAL ASPECTS OF DRIVING OSCILLATIONS

Using the data presented thus far as background material, it is now possible to discuss the theoretical considerations leading to a postulation of the mechanism of oscillation. It will be shown how Rayleigh's criterion[4] can be applied to these oscillations and what components of the heating unit are involved in the generation of oscillations. In connection with the discussion of the mechanism of oscillation, the separate effects of primary air flow, firing rate, and furnace volume on the amplitude of oscillation will be brought out.

RAYLEIGH'S CRITERION

One of the basic hypotheses which explain oscillations in combustion systems was first proposed by Lord Rayleigh[4]. Appendix A has been devoted to a fuller explanation of his criterion. For this discussion, however, it is sufficient to state that an oscillation can be sustained when the rate of heat release of a flame is a maximum when the pressure of the system is a maximum or, conversely, when the rate of heat release of the flame is a minimum when the pressure is a minimum. If one or the other of these conditions is met and the driving force exceeds the damping losses, then an oscillation will be sustained. It should be noted that 2 events are involved in this criterion; there must be some means by which a periodic pressure fluctuation is present at the flame and there must be a periodic variation in the rate of heat release of the flame. Applying these conditions to a gas-fired single-port heating unit, it will be shown that the required pressure fluctuations arise from a Helmholtz-type oscillation of the volume of gases in the furnace or boiler into and out of the inlet openings and the stack. The periodic variations in the rate of heat release occur as a result of vortex shedding from the burner lip.

PRESSURE VARIATION IN HEATING UNIT

Combustion of a turbulent gaseous mixture at and beyond the burner lip is a partially random process and as such would produce random noise. However, because the burner is contained within a volume in the furnace or boiler, the burner will be in an environment having one or more resonant frequencies. Therefore, although the flame produces noise at many frequencies, those frequencies which are equal to the resonant frequencies of the furnace or boiler will be accentuated. Thus, a standing wave of fluctuating amplitude will be set up inside the furnace or boiler. The internal volume of most drum-type furnaces, for instance, is such that the Helmholtz resonant frequency will fall in the range of frequencies between 40 and 80 cps. It is this mode of oscillation that is of major concern to this discussion.

In the subsequent discussion, the resonant frequency of the heating unit will be assumed as constant for any given configuration. The resonant frequency is a function of the volume of the combustion chamber, the configuration of the inlet region of the heating unit, and the absolute temperature of the gases in the stack. These items varied during the course of the different investigations; thus, the resonant frequency also varied. However, this variation was believed to be small and was considered negligible in the following discussion.

POSSIBLE SOURCES OF DRIVING

Periodic variations in the rate of heat release required to sustain oscillations can be provided by vortices passing through the flame. The cause of these vortices, however, has to be understood before the mechanism of oscillation can be fully determined.

The presence of these vortices could be explained by 2 different mechanisms. It was possible that a pressure oscillation in the furnace would cause a pressure oscillation of the same frequency to occur in the burner tube. Whereas the pressure amplitude at the spud would be essentially zero, the pressure at the burner outlet would correspond to that of the furnace. Therefore, there would also be a velocity fluctuation at the burner outlet. During part of a complete velocity fluctuation cycle, the mixture would emerge from the burner in the form of a jet whereas during a later portion of the cycle, under severe oscillating conditions, the gases would flow back into the burner much like water flowing coreless-fashion into a drain. This fluctuation in velocity could cause vortices to shed periodically from the burner lip. The frequency of these vortices would be the resonant frequency of the furnace with the magnitude governed by the length of the burner tube as well as the amplitude of oscillation in the furnace. The second possibility suggested was that the vortex shedding was an aerodynamic rather than an acoustic phenomenon. Thus, it would be dependent only on the burner port diameter and the average combustible gas velocity.

The first mechanism was found not to be the principal cause of the vortices when more extensive studies were made of the effect of changes in burner-tube length on the amplitude of oscillation (Appendix B). However, as will be shown later, this effect does seem to be present in a small way, and may tend to regulate the shedding of the vortices at high primary air-flow rates. By taking high-speed moving pictures of the flame, with the burner outside of the furnace, it was found that the vortices still were present and that, in the absence of a furnace with its own resonant frequency, their frequency was a function of the combustible gas velocity. Once this fact had been established, the mechanism could be postulated.

AERODYNAMIC VORTEX SHEDDING AS A SOURCE OF DRIVING

Vortex shedding from objects immersed in fluid streams is not an uncommon occurrence in nature[5]. One way to determine if a vortex shedding is of the aerodynamically induced type is to perform a correlation of 2 dimensionless parameters which involve the frequency of the disturbance, the physical size of the object, and several flow properties of the fluid. For this correlation the Strouhal number ($\nu D/V$) is plotted as a function of Reynolds number ($\rho VD/\mu$). The terms in these dimensionless groups are defined as follows:

ν = frequency of the disturbance, cycles per second.
D = diameter of the object, feet.
V = average velocity of the fluid, feet per second.
ρ = density of the fluid, pounds per cubic foot.
μ = absolute viscosity of the fluid, pounds per foot per second.

In this particular correlation, ν is the average frequency of vortex shedding obtained from a frame-by-frame analysis of the high-speed moving pictures, D is the burner-port diameter, and V is the average velocity of the combustible mixture

flowing out of the burner port. ρ and μ were found for the gas and air mixtures at the conditions when the moving pictures were taken.

Fig. 4 is a log-log plot which shows the results of this correlation when applied to 2 independently run sets of data for the single-port gas burner outside of the furnace. The values used to obtain the data points are presented in Table B-1 of Appendix B. The conditions under which each set of data was obtained were duplicated as well as possible. The 3 high points could represent errors in determining the vortex-shedding frequencies or, more likely, vortex shedding at double its fundamental or normal frequency as indicated by the dashed line. Such a phenomenon has been reported occasionally in the literature. The remainder of the data scatter by only

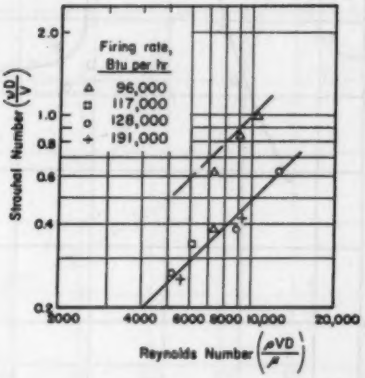


FIG. 4. . . . STROUHAL NUMBER AS A FUNCTION OF REYNOLDS NUMBER FOR THE SINGLE-PORT BURNER EXTERNAL TO THE EXPERIMENTAL FURNACE

a small amount about the solid line with a one-to-one slope. It is felt that this correlation is significant, and indicates that the frequency of vortex shedding is dependent principally on the average velocity of the combustible gas flowing out of the burner tube, and to a lesser extent, if any, on the burner port diameter. It should also be noted that firing rate, which was varied widely for these studies, did not affect the correlation, except that a low-firing rate and high Reynolds number seemed to provide the second mode frequency of vortex shedding.

The production of vortices from the end of a tube through which a fluid is passing has been reported by Wille, Domm, and Wehrman[6]. They present results on vortex shedding from the lip of a nozzle through which a fluid is passing at low velocities. This report gives a confirmation to the phenomenon of vortex shedding observed during the experimental work, and to the general relation observed between the dimensionless groups.

After the vortices shed from the burner lip, they pass through the flame front where they momentarily change the rate of heat release of the flame. Since the

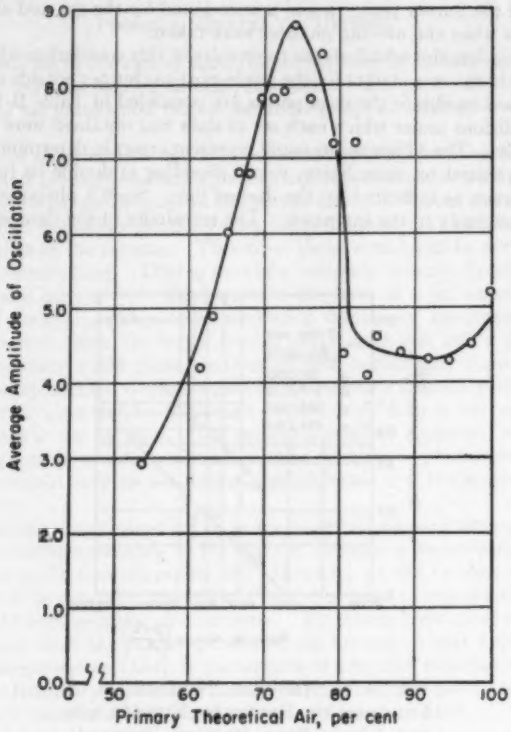


FIG. 5. . . .AVERAGE AMPLITUDE OF OSCILLATION AS A FUNCTION OF PRIMARY THEORETICAL AIR FOR THE SINGLE-PORT GAS FURNACE AT A FIRING RATE OF 191,000 BTU PER HR INPUT

passage of vortices is periodic, it can be expected that the variation in the rate of heat release will likewise be periodic.

EXPERIMENTAL VERIFICATION OF THE MECHANISM OF OSCILLATION

As has been shown, the oscillation present in gas-fired single-port heating units results from a pressure fluctuation and a variation in the rate of heat release. It is the coupling or phasing between these components which determines the amplitude of the oscillation. Obviously, when the coupling is precise, the amplitude will be a maximum. This will occur when the normal frequency of vortex shedding is equal to the resonant frequency of the furnace or boiler. An oscillation of reduced am-

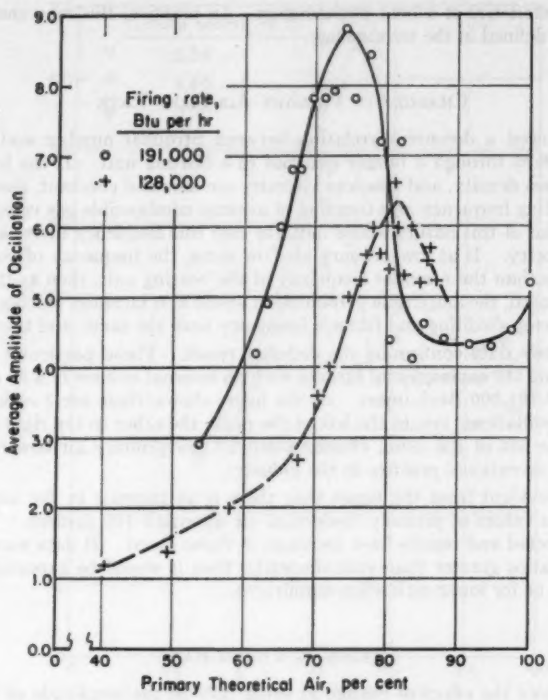


FIG. 6. . . .AVERAGE AMPLITUDE OF OSCILLATION AS A FUNCTION OF PRIMARY THEORETICAL AIR FOR A FURNACE VOLUME OF 4.66 CU FT AND 2 FIRING RATES

plitude may still be sustained when the normal frequency of vortex shedding is either less than or greater than the resonant frequency of the heating unit.

If the 2 frequencies were distinct and different, and no interaction took place, then this difference in frequency would result in a beat phenomenon. However, the frequencies are not distinct. Both the resonant furnace frequency and the normal frequency of vortex shedding are *sloppy*. Furthermore, recent work by Garber[7] has shown that in an axially symmetric vortex-shedding system, the presence of a driving frequency will, in some manner, help time the vortex shedding. By Rayleigh's criterion, the proper input of heat resulting from the vortex shedding will drive the acoustic oscillation. Thus, the 2 oscillating systems couple or interact, with a feedback system set up. However, one still expects that as the frequency of the 2 basic components draw apart, the amplitude of oscillation involved in the system will decrease. Furthermore, at some point it will be expected that the feedback system will break down, and the oscillation will take on more and more

of the characteristics of a beat phenomenon. In practice, this may show up as a surging as is defined in the terminology.

CHANGES IN PRIMARY AIR-FLOW RATE

Fig. 4 showed a definite correlation between Strouhal number and Reynolds number for flow through a burner external to a heating unit. If the burner port diameter, mass density, and absolute viscosity are assumed constant, then a plot of vortex-shedding frequency as a function of average combustible gas velocity can be made. A plot of this nature would indicate that this frequency increased with increasing velocity. If at low primary air-flow rates, the frequency of vortex shedding was less than the resonant frequency of the heating unit, then as the primary air was increased, the amplitude of oscillation would first increase, reach a maximum when the vortex shedding and furnace frequency were the same, and then decrease.

Fig. 5 shows data confirming the deduced result. These particular data were obtained using the experimental furnace with an internal volume of 4.66 cu ft and a firing rate of 191,000 Btuh input. As the figure shows, there are 2 regions of low-amplitude oscillations; one to the left of the peak, the other to the right. It is the region to the left of the peak, characterized by low-primary air-flow rate, which represents conventional practice in the industry.

It is also evident from the figure that there is an increase in the amplitude of oscillation as values of primary theoretical air approach 100 percent. This effect is to be expected and results from increases in flame speed. If data were obtained at air-fuel ratios greater than stoichiometric, then it would be expected that the trend would be for lower oscillation amplitudes.

CHANGE IN FIRING RATE

Fig. 6 shows the effect of change in firing rate on the amplitude of oscillation. For a furnace volume of 4.66 cu ft, the 2 firing rates were 128,000 Btuh and 191,000 Btuh input. As can be seen from the figure, the curve for the higher burning rate reaches a maximum at a lower value of percent primary theoretical air than does the curve for the lower firing rate. This occurs because the quantity of air required to obtain the same percent primary theoretical air is greater for the higher firing rate than it is for the lower. Thus, the velocity of combustible gas flow is greater. Since the frequency of vortex shedding is a function of this velocity, the condition where the vortex-shedding frequency is equal to the resonant frequency of the heating unit is reached at a lower percent of primary theoretical air for the higher firing rate than for the lower. The higher amplitude for the 191,000 Btuh firing rate results from the increased energy available for driving.

CHANGES IN HEATING UNIT VOLUME

Fig. 7 is presented to show the effect of changes in heating-unit volume on the amplitude of oscillation. The firing rate for these data was 191,000 Btuh input. These changes in volume were accomplished by inserting closed cylinders of various volumes inside the experimental furnace. As this figure shows, the maximum amplitude is shifted to higher values of percent primary theoretical air for smaller net furnace volumes. Decreasing the furnace volume will raise the resonant fre-

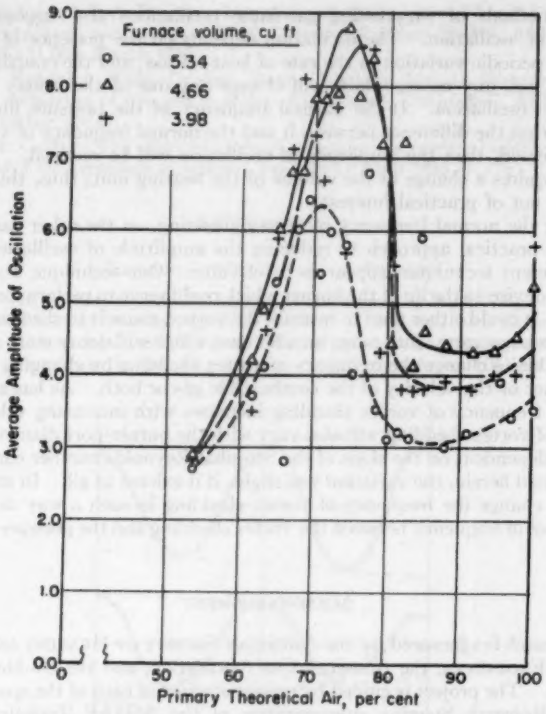


FIG. 7. . . .AVERAGE AMPLITUDE OF OSCILLATION AS A FUNCTION OF PRIMARY THEORETICAL AIR, THREE FURNACE VOLUMES AND A FIRING RATE OF 191,000 BTU PER HR INPUT

quency of the furnace and thus require a higher rate of primary air to achieve a vortex-shedding frequency equal to the resonant frequency of the furnace.

CONCLUDING REMARKS

It has been shown that the mechanism of oscillation in single-port gas-fired heating units involves a vortex-shedding phenomenon at the lip of the burner tube. The presence of periodic vortices passing through the flame front produces a periodic change in the rate of heat release of the flame. At the same time, there is a pressure oscillation in the volume of the heating unit whose frequency, in the absence of driving effects, would be equal to the frequency of the unit. The coupling which exists between the rate of heat release of the flame and the natural pressure oscillation governs the amplitude of oscillation produced by the heating unit.

Several methods of suppressing gas-flame oscillations are suggested by the mechanism of oscillation. The oscillation depends on the presence of a pressure fluctuation, periodic variation in the rate of heat release, and the coupling between the two. Thus, any means which will change any one of the 3 may reduce the amplitude of oscillation. If the natural frequency of the pressure fluctuation is changed so that the difference between it and the normal frequency of vortex shedding is increased, then the amplitude of oscillation will be reduced. To do this, however, requires a change in the volume of the heating unit; thus, this technique is normally not of practical interest.

Changing the normal frequency of vortex shedding, on the other hand, offers a much more practical approach to reducing the amplitude of oscillation. In this light, 2 different techniques appear to be of value. One technique would involve attaching a device to the lip of the burner which could serve to perform one of several functions. It could either tend to restrain the vortex, cause it to shed continuously, or break the large vortex into many smaller ones which will decay more rapidly. It is also possible to change the frequency of vortex shedding by changing the burner-port diameter or the velocity of the combustible gas or both. As has already been stated, the frequency of vortex shedding increases with increasing velocity. The frequency of vortex shedding will also vary with the burner-port diameter, but how it varies is dependent on the slope of the Strouhal-Reynolds number curve. In the data presented herein, the variation was slight, if it existed at all. In any case, it is possible to change the frequency of vortex shedding in such a way as to increase the difference of frequency between the vortex shedding and the pressure fluctuation.

ACKNOWLEDGMENT

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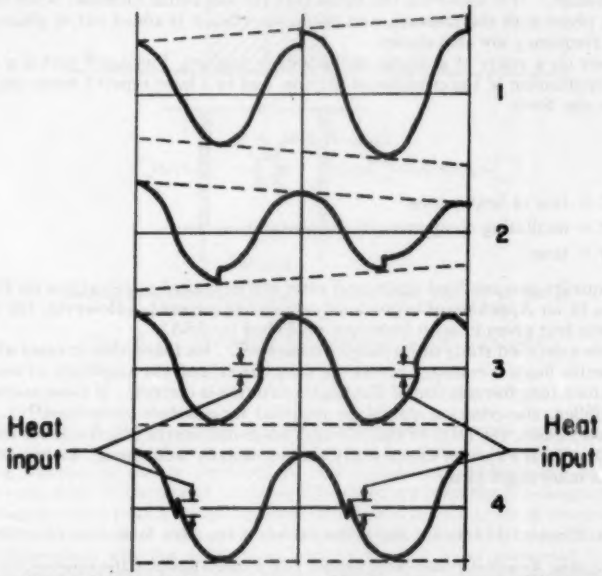
APPENDIX A

RAYLEIGH'S CRITERION OF DRIVING

Rayleigh[4] was the first to advance an hypothesis regarding the driving of oscillations by a periodic release of heat in a gaseous medium. To quote, with minor changes in terminology, "If heat is periodically added to, and taken from, a mass of gas oscillating, for example, in a cylinder bounded by a piston, the effect produced will depend upon the



Phase of heat supply relative to phase of pressure	Effect on amplitude	Effect on frequency
1 In phase	Increase	None
2 Opposite phase	Decrease	None
3 Quarter-period before	None	Increase
4 Quarter-period after	None	Decrease

FIG. A-1. . . .EFFECT ON OSCILLATION BY PERIODIC SUPPLY OF HEAT (AFTER WOOD; FIG. 3.10, P. 93 OF REFERENCE 8)

phase of the oscillation at which the transfer of heat takes place. If heat is given to the gas at the moment of highest pressure, or taken from it at the moment of lowest pressure, the oscillation is encouraged. On the other hand, if heat be given at the moment of lowest pressure, or abstracted at the moment of highest pressure, the oscillation is discouraged. When the transfer of heat takes place at the moment of either highest or lowest pressure, the frequency is not affected. If the gas is at its normal density at the moment when the transfer of heat takes place, the oscillation is neither encouraged nor discouraged, but the frequency is altered. Thus, the frequency is raised if heat is added to the gas at a quarter period before the phase of highest pressure; and the frequency is lowered if the heat is added a quarter period after the phase of highest pressure."

"In general, both kinds of effects are produced by a periodic transfer of heat. The frequency is altered, and the oscillation is either encouraged or discouraged. But there is no effect of the second kind if the gas concerned be at a pressure node, nor if the transfer of heat be the same at any stage of lower pressure as at the corresponding stage of high pressure."

Fig. A-1 presents an interesting method advanced by Wood* for physically picturing Rayleigh's explanation. Wood treats the increment of heat as a corresponding increment of pressure. It is seen from the figure that the amplitude increases when the heat is added in phase with the pressure, and decreases when it is added out of phase. The changes in frequency are also shown.

In a report on a study of acoustic oscillations in burners, Putnam** stated a mathematical formulization of the criterion of driving, and in a later report† made use of this criterion in the form

$$\int_{\text{cycle}} H\,p\,dt > 0,$$

where

H = rate of heat release

p = oscillating component of the pressure

t = time.

This inequality was outlined again, and some mathematical justifications for the form were given, in an Appendix of a previously-published paper††. However, the explicit equation was first given in open literature published by ASA‡.

Chu made a detailed study of Rayleigh's criterion◘. He found that in cases where the ratio of specific heats is constant across the flame front, and the amplitude of oscillation is low, the foregoing formulation of Rayleigh's criterion is correct. If these assumptions are not fulfilled, the criterion should be modified for absolute correctness◘, +. However, for most cases, the form of the criterion presented herein is suitable for use, since the conditions in the system under study are not usually well enough known to justify the use of a more exact form.

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* A. Wood: Chapter III (*Acoustics*, Interscience Publishers, Inc., New York, 1945, 1st ed. 2d reprint, p. 92).

** A. A. Putnam: *Acoustical Oscillations in Burners*, First Progress Report to Headquarters, Air Materiel Command (Contract W 33-038ac-14202, Project MX-833, from Battelle Memorial Institute, October 1946).

† A. A. Putnam: *Acoustic Oscillations in Burners*, Eighth Progress Report to Headquarters, Air Materie Command (Contract W 33-038ac-14202, Project MX-833, from Battelle Memorial Institute, May 1947).

†† A. A. Putnam and W. R. Dennis: A study of burner oscillations of the organ-pipe type (*ASME Transactions*, Vol. 75, 1953, p. 15).

‡ A. A. Putnam and W. R. Dennis: Burner oscillations of the gauze-tone type (*Journal of the Acoustical Society of America*, September 1954, p. 716).

◘ Boa-Teh Chu: *Stability of Systems Containing a Heat Source—The Rayleigh Criterion* (National Advisory Committee for Aeronautics, RM 56D27, June 26, 1956).

+ B V Raushenbakh: A remark by Rayleigh concerning thermal sound excitation (*Doklady Akademii Nauk*, SSSR, Vol. 91 (4), 1953, p. 743. Also *Applied Mechanics Reviews*, No. 1208 (8), April 1955).

Kovasnay[9] outlined a rather clear physical basis for the Rayleigh criterion of driving as follows:

"Consider a tube of finite length. Divide the tube into many fictitious compartments (say of equal size), in one of which heat is being added periodically from an external source. This is illustrated in Fig. A-2 where the compartments are separated by planes (shown as dashed lines in the figure), while the shaded compartment bounded by the planes marked A and A′ is the one in which heat is being added. In the absence of viscosity and heat conductivity, there is no less of generality by replacing the fictitious planes A and A′ by 2 solid plane surfaces (or *pistons*) provided that these walls always move in synchronism with the motion

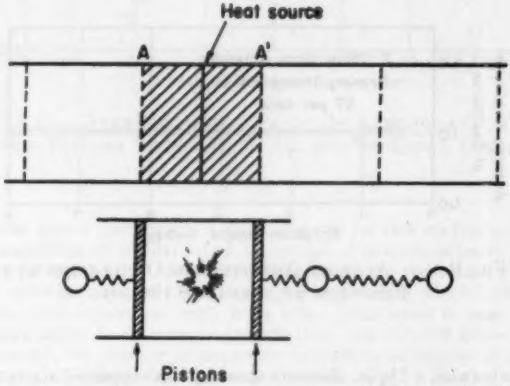


FIG. A-2. . . .SKETCH SHOWING ENGINE ANALOGY

of the fictitious planes A and A′. The other compartments serve, as is well known, as a mechanical spring-mass system (or flywheels in the engine analogy), where mechanical energy produced by the *engine* compartment can be stored. The work done by the engine can be easily calculated from the PV-diagram. Depending on when heat is added and subtracted during each cycle of vibration, the amount of mechanical work produced is greater than, equal to, or less than zero; this determines whether mechanical energy is given to or extracted from the spring-mass system (*i.e.*, the flywheels). For the case when the amount of heat energy released by that heat source depends on the magnitude of fluctuation in the system and is zero when there is absolutely no fluctuation in the system, the initial disturbances act as an *engine starter*. It is at once clear from the PV-diagram that if heat is added when the pressure in the engine is the highest and taken away when the pressure in the engine is the lowest, the maximum amount of mechanical work is obtained. Furthermore, it is also clear from the engine analogy that the real criterion of amplification of a disturbance is that the net mechanical work done by the engine per cycle must be greater than the loss through viscous dissipation and, hence, must at least be greater than zero."

To summarize, the addition of heat in a periodic manner to a body of gas can result in the driving of an acoustical oscillation. The component of heat release in phase with the pressure serves to overcome damping forces or to increase the amplitude of oscillation, while the component of heat release out of phase with the pressure changes the frequency of this oscillation.

APPENDIX B

SINGLE-PORT BURNER

Description: A conventional mono-port burner was used for all of the studies discussed in this paper. The port diameter was 2 in. A ⅜-in. axial rod supported the spreader.

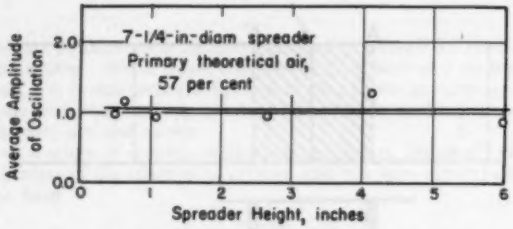


FIG. B-1. . . .AVERAGE AMPLITUDE OF OSCILLATION AS A FUNCTION OF SPREADER HEIGHT

Unless noted otherwise, a 7¼-in. diameter spreader was supported above the burner at a minimum distance from the burner lip of 2½ in.

For the early studies, an open spud was used with the primary air supplied from the surrounding atmosphere. Natural gas was fed from the city lines into a 600 cu ft capacity gas holder. This gas holder then supplied the burner with fuel gas at a constant pressure. The natural gas was metered using a rotameter. The percentage of primary air by volume in the air-fuel gas mixture was determined by drawing off a sample from near the downstream end of the burner tube, and passing the sample through a relative thermal conductivity meter. For most of the parameter studies, however, a 1-cu ft box enclosing the spud was used. Compressed air was passed through a critical flow orifice and into the box. By metering both the air and fuel gas, it was possible to determine the percent primary theoretical air directly.

A number of different *burner parameters* were studied during the course of this investigation to determine what effect each had on the amplitude of oscillation. For these studies, a firing rate of approximately 96,000 Btuh input was used.

Fig. B-1 shows the effect of *spreader height* on the average amplitude using the 7¼-in. diameter spreader and 57 percent theoretical primary air. This plot indicates that, within the scatter of the data, the spreader height can be said to have no effect on the amplitude of oscillation. It will be noted that in this test, heights down to ½ in. were used. Attempts to decrease this height still further were impossible because the flame became unstable and went out.

Fig. B-2 indicates the amplitude of oscillation as a function of percent primary theoretical air and *spreader diameter*. For this study, 3 spreader diameters were used; namely, 5½ in., 7¼ in., and 10 in. The narrow scatterband of the data indicates that the spreader diameter has little or no effect on the amplitude of oscillation.

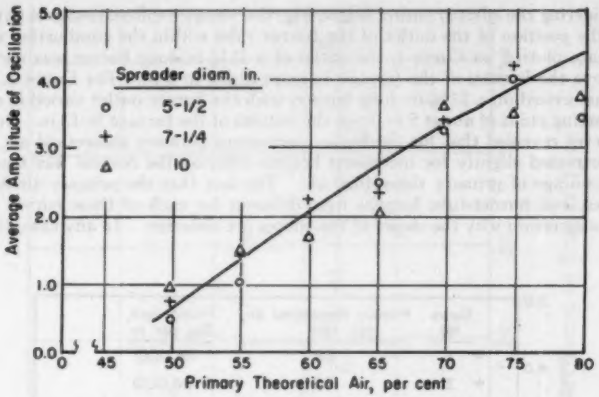


FIG. B-2. . . .AVERAGE AMPLITUDE OF OSCILLATION AS A FUNCTION OF PRIMARY THEORETICAL AIR AND SPREADER DIAMETER

Another of the burner parameters is the *firing rate for rich air-fuel ratios.* Fig. B-3 indicates the amplitude of oscillation as a function of percent primary theoretical air and 3 different values of firing rate. As can be seen from the figure, the scatterband of the data is quite narrow and there is a trend in the data which indicates that the amplitude of oscillation increases with firing rate. This trend is more noticeable on Fig. 6 of the text where firing rates of 128,000 Btuh and 191,000 Btuh are compared. As stated previously, the increase in amplitude basically results from an increase in the amount of energy available for driving the oscillations.

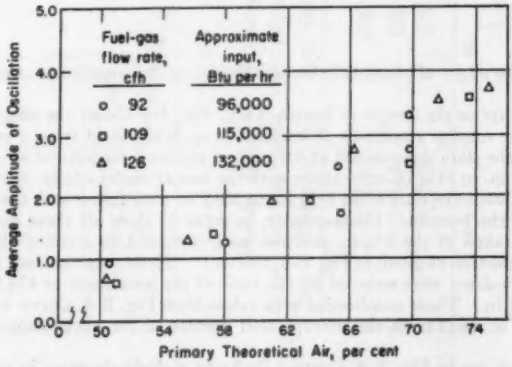


FIG. B-3. . . .AVERAGE AMPLITUDE OF OSCILLATION AS A FUNCTION OF PRIMARY THEORETICAL AIR AND FIRING RATE

In considering the *effect of burner height*, Fig. B-4 shows 2 different sets of data on the effect of the position of the outlet of the burner tube within the combustion chamber. For the data plotted on Curve 1, the outlet of a 21¼-in.-long burner was varied from 13 in. above the bottom of the furnace in steps down to 6 in. For Curve 2, the procedure was revised on a 22½-in.-long burner, with the burner outlet varied in elevation from a starting point of about 5 in. from the bottom of the furnace to 13 in. Inspection of the curves revealed that for the higher percentage primary theoretical air, the amplitude decreased slightly for increasing heights whereas the reverse was true for the lower percentage of primary theoretical air. The fact that the primary air-flow rates, firing rates, and burner-tube lengths were different for each of these curves makes it difficult to ascertain why the slopes of the curves are different. In any case, the results

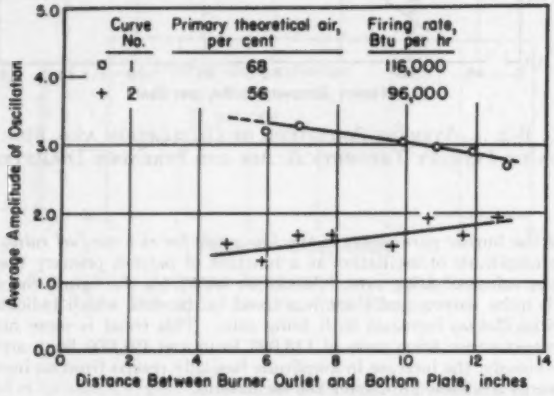


FIG. B-4. . . .EFFECT OF POSITION OF BURNER OUTLET ON THE AMPLITUDE OF OSCILLATION

indicate that the effect of changes in burner height on the amplitude of oscillation will be small.

Effect of Change in the Length of Burner Tube: Fig. B-5 shows the effect of length of the burner tube on the amplitude of oscillation as determined from 2 separate series of tests. For the data determined at 68 percent primary theoretical air, Curve 1, the data from 14¼ in. to 21¼ in. were taken with the burner outlet 4⅞ in. above the bottom of the furnace, while the data from 22¼ in. to 30⅛ in. were taken with the burner outlet 13¼ in. above the bottom. Consequently, in order to show all these data on a single plot, the data taken at the 4⅞-in. position were corrected to account for the effect of burner-outlet position as given in Fig. B-4, Curve 1. To do this, the data for the 4⅞-in. burner-outlet position were reduced by the ratio of the amplitude at 4⅞ in. to the amplitude at 13¼ in. These amplitudes were taken from Fig. B-4, Curve 1, with the amplitude at 4⅞ in. read from the extrapolated portion of the curve shown as a dashed line.

The results shown in Fig. B-5, Curve 1, indicate a slight decrease in amplitude with increase in burner-tube length for lengths between 13¼ in. and 21¼ in. Superimposed on the change in amplitude is an apparent wave in plotted points resulting, it is believed, from a chance scatter data.

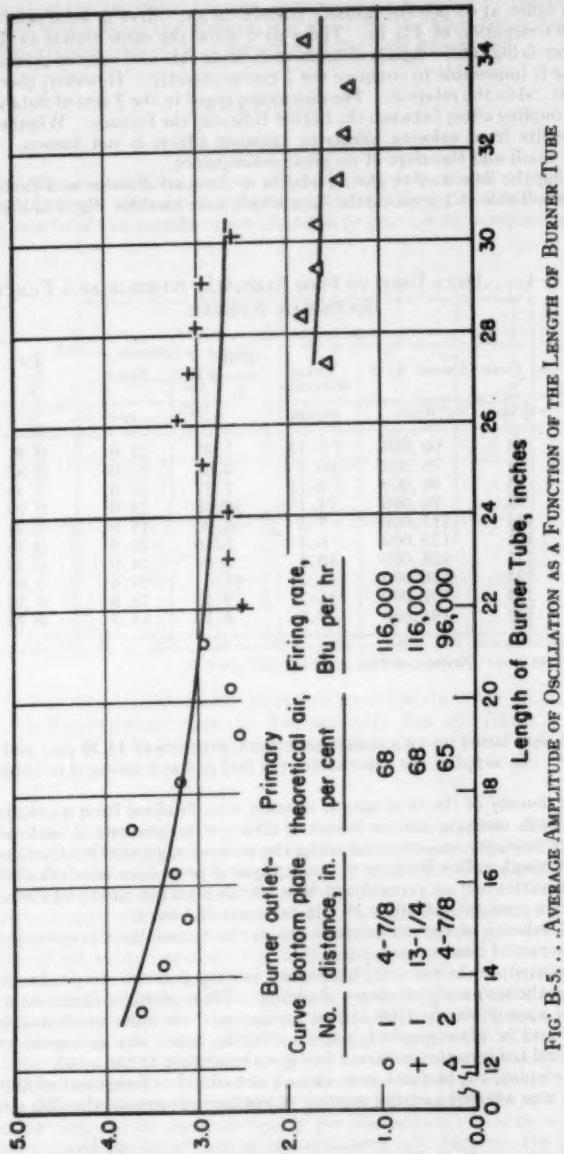


FIG. B-5. . . AVERAGE AMPLITUDE OF OSCILLATION AS A FUNCTION OF THE LENGTH OF BURNER TUBE

The data taken at 65 percent primary theoretical air, Curve 2, were determined for a burner-outlet elevation of 4⅞ in. These data show the same trends as did the data for the higher firing rate. Again, differences in firing rate and percent primary theoretical air make it impossible to compare the 2 curves directly. However, there is a slight downward trend in the relation. The downward trend in the 2 sets of data is indicative of a slight coupling effect between the burner tube and the furnace. Whether this slight coupling results from velocity effects or pressure effects is not known. The effect, however, is small and therefore of no great consequence.

In studying the *data used to plot correlation of Strouhal Number as a function of Reynolds Number*, Table B-1 presents the data which were used for Fig. 2 of the text. The

TABLE B–1. . . . DATA USED TO PLOT STROUHAL NUMBER AS A FUNCTION OF REYNOLDS NUMBER

AIR FLOW	GAS FLOW	FIRING RATE	TOTAL VELOCITY	$\frac{\rho VD}{\mu} \times 10^{-4}$	FREQ.	$\frac{vD}{V}$	SYMBOL
FT³/HR	FT³/HR	BTUH	FT/SEC	—	CPS	—	—
595	91	96,000	8.75	7.37	32.0	0.61	△
730	91	96,000	10.5	9.04	53.0[a]	0.83	△
595	91	96,000	8.75	7.37	20.0	0.38	△
860	91	96,000	12.1	10.55	72.0	0.99	△
463	111	117,000	7.3	6.06	15.0	0.34	□
371	121	128,000	6.3	5.16	10.0	0.26	0
705	121	128,000	10.5	8.84	24.0	0.38	0
1035	121	128,000	14.7	12.6	54.6	0.62	0
699	181	191,000	11.2	9.25	28.0	0.42	+
367	181	191,000	7.0	5.50	10.5	0.25	+

[a] Average value used—Diameter = 2 in.

calculations were based on an average barometric pressure of 14.30 psia and a temperature of 80 F. To simplify the calculation, the fuel gas was assumed to consist solely of methane.

The mass density of the combustible mixture was obtained from a calculation of the densities of both methane and air based on their partial pressure at each mixture rate. The absolute viscosity was obtained using the method suggested by Uyehara and Watson[□□]. Although values for both of these physical properties were calculated for each point, the densities had an average of 2.06×10^{-3} lb mass per cu ft and the absolute viscosities had an average of 0.408×10^{-6} lb mass per (ft) (sec).

The total velocity of the combustible gas at the burner lip was calculated from the total volume rate of flow for each point.

As was pointed out in the text, high-speed moving pictures were taken of the flame to determine the frequency of vortex shedding. These pictures were taken at an average speed of approximately 1000 frames per second. In order to illuminate the flame so that it could be photographed, sodium chloride vapor was generated externally to the burner and fed into the premixed fuel gas-air mixture at the spud.

After developing, the pictures were viewed and edited. A segment of film 200 to 300 frames long was selected and the number of vortices present on the film strip counted.

□□ O. A. Uyehara and K. M. Watson: A universal viscosity correlation (*Technical Section, National Petroleum News* October 1944, p. R-714).

A timing signal at the margin of the film, automatically recorded by the camera, provided means of determining the average film speed for the film strip. From this information, the average frequency of vortex shedding was determined.

APPENDIX C

EXPERIMENTAL FURNACE

Description: The experimental furnace shown in Fig. 2 contained a 14-in.-diameter, 5-ft-long, constant area, combustion chamber. A wrap-around heat exchanger covered about three fourths of the outside of the chamber to prevent its overheating. A draft

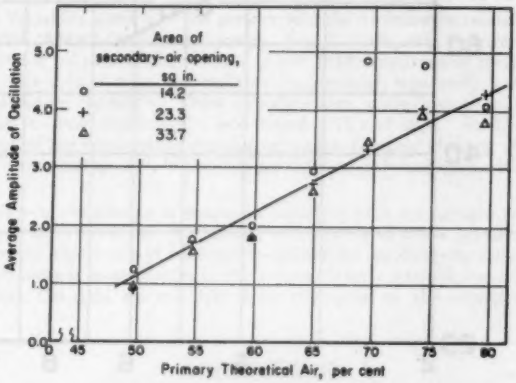


FIG. C-1. . . .AVERAGE AMPLITUDE OF OSCILLATION AS A FUNCTION OF PRIMARY THEORETICAL AIR AND AREA OF SECONDARY-AIR OPENING

diverter was placed in the stack within 6 in. of the top of the furnace. A shutter mechanism at the bottom of the furnace allowed regulation of the secondary air flow.

Variables Studied: Several furnace variables were studied. These will now be discussed with the first being *effect of secondary-air distribution.*

For the initial studies, the secondary-air inlet at the bottom of the combustion chamber had been covered by a fine mesh screen, of about the same porosity as ordinary window screen, which served to smooth out the flow of secondary air. The first modification of the furnace in the investigation of the effect of secondary-air distribution was to replace the fine screen by 2 layers of coarse screen, standard ¼-in.-mesh hardware cloth, with the top layer rotated so that the squares of the screen were aligned at an angle of about 45 deg to one another. The second modification consisted of 3 flat annular rings of different outer diameters, which could be placed in various combinations on the screen around the burner tube and thus provide a means for channeling secondary-air flow in various ways. The rings were of equal width, and were sized so that (1) the outer diameter of the outer ring was the same as that of the combustion chamber, and (2) the 3 rings together covered the entire area of the secondary inlet between the burner tube and the wall.

In the experiments made with the foregoing modifications to the secondary-air inlet, oscillations of appreciable amplitudes were generated at reasonable CO_2 concentrations with primary mixtures containing no more than 70 percent theoretical air. The amplitude of oscillation did not change significantly as the channeling of the secondary air was changed by first leaving the annular rings out and then installing the rings in various combinations, such as the outer and intermediate, the inner and intermediate, and so on.

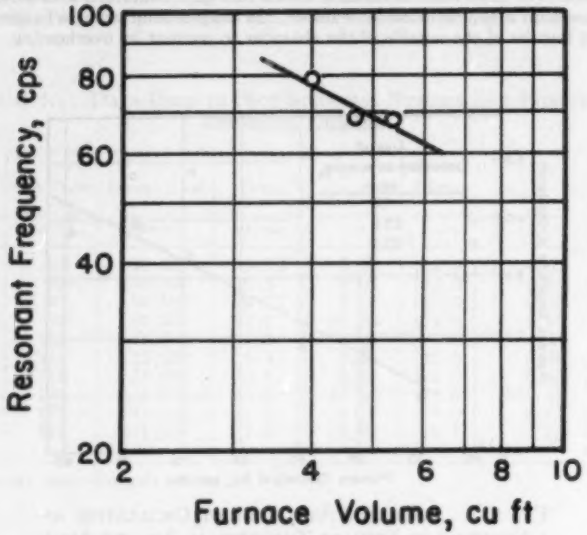


FIG. C-2. . . .FURNACE RESONANT FREQUENCY AS A FUNCTION OF NET FURNACE VOLUME FOR THE EXPERIMENTAL FURNACE OPERATING AT A LOW PRIMARY AIR FLOW

In studies discussed from this point on, the outer and intermediate rings were used together, with the inner ring omitted.

In studying the *effect of changes in secondary-air opening*, the effect on the amplitude of oscillation resulting from changes in the area of the secondary-air opening is shown by Fig. C-1. It appears that there is a slight effect present in that the smallest area gave consistently the highest amplitude of oscillation. However, the effect is definitely small when compared to changes in the amplitude of oscillation resulting from changes in the percent primary theoretical air. The slight change in the amplitude resulting from changes in the secondary-air opening are felt to be caused by changes in the acoustical system rather than by changes in the rate of flow of secondary air. Decreasing the area of the opening would shift the position of the pressure antinode in the furnace; in this case, toward the flame.

In considering the *effect of furnace and stack temperature*, it can be shown by theoretical considerations that the temperature of the gases in the combustion chamber does not

affect the resonant frequency of the furnace. This frequency, however, is somewhat sensitive to changes in the temperature of the stack gases. It has been shown that the amplitude of oscillation is dependent on the furnace resonant frequency and therefore would be dependent upon the stack temperature to some degree. This would help to explain the phenomenon that some furnaces are noisy when cold and quiet when hot or conversely.

The *effect of changes in atmospheric conditions* on the amplitude of oscillation has not been studied as a separate parameter. It is not possible to accredit any particular oscillation phenomenon to changes in air pressure, temperature, or humidity of the magnitude normally occurring in one locality. The density and acoustic velocity would be affected by changes in barometric pressure and temperature. They, in turn, could change the amplitude of oscillation. Again, however, the change should be small. A large change in altitude could cause a sufficient change in the density of the air to affect the flow rate of the primary air. This, in turn, could cause a unit to be noisier.

Among the variables studied in the project was the *variation in resonant frequency of the Furnace with changes in furnace volume*. Fig. 7 of the text indicated the effect of changes in furnace volume on the position of the peak amplitude of oscillation. These changes in furnace volume were accomplished by inserting, separately, 2 closed cylinders inside the combustion chamber. These cylinders were made from 10-in. stove pipe and were 15 and 30 in. long, respectively, and closed with end caps. Each cylinder was 8 in. from the top of the combustion chamber so as not to cause choking of the flue gases or to change the percentage CO_2 in the flue gas from what it had been without the cylinder in place.

Fig. C-2 indicates the change in resonant frequency with net furnace volume. These frequencies were determined for the furnace when firing at a low primary-air flow rate in order to prevent high levels of combustion noise from masking the induced sound signal. Although there is some scatter in the points about a straight line with the correct theoretical slope, the data are still felt to be indicative of the changes in resonant frequency.

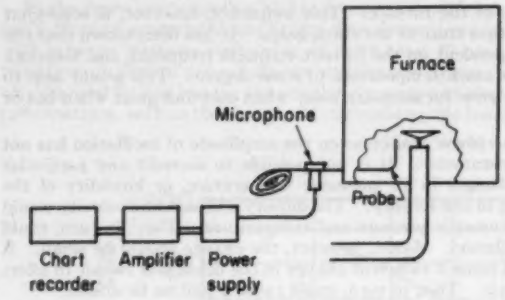
APPENDIX D

INSTRUMENTATION

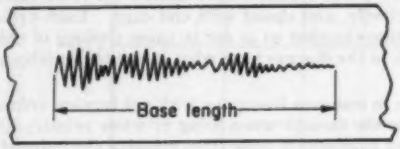
Subfigure (a) of Fig. D-1 indicates schematically the instrumentation used to measure the amplitude of oscillation for these studies. A probe microphone was used to measure the sound pressure inside the furnace. The probe was known not to affect the frequency of the noise input received by the microphone but did attenuate the amplitude. The microphone was a commercial condenser-type microphone which had a frequency cutoff at 20 cps. The amplifier was used to increase the voltage signal from the microphone to a high enough level to drive the chart recorder. The recorder produced a permanent record of the voltage output of the microphone on a strip of paper which traveled at a constant speed. The entire system had an over-all frequency response of 20 to 60 cps.

Subfigure (b) of Fig. D-1 indicates a typical noise trace obtained from the furnace during operation. To determine the amplitude of oscillation, a base length of 200 mm was chosen in a region of record which typified the particular data. Using a rolling-type planimeter, an area made up of the upper and lower envelopes of the noise trace over the chosen base length was determined. This area was divided by twice the base length and multiplied by the planimeter constant to obtain an average amplitude.

The underlying principle behind the *determination of the resonant frequency of a furnace* is that the sound level in the furnace is greatest when the frequency of the sound being fed into the furnace is the resonant frequency of the furnace. Fig. D-2 shows schematically the instrumentation used. The sound source consisted of a loudspeaker driven by an amplifier. An audio oscillator produced the initial signal of known frequency. The probe microphone was used to measure the sound pressure inside the furnace. The



(a) Schematic Sketch of Instrumentation Used in Measuring the Amplitude of Oscillation



(b) Sketch of Typical Noise Trace From Chart Recorder

FIG. D-1. . . .SKETCHES OF INSTRUMENTATION USED IN DETERMINING AMPLITUDE OF OSCILLATION IN THE EXPERIMENTAL SINGLE-PORT GAS FURNACE

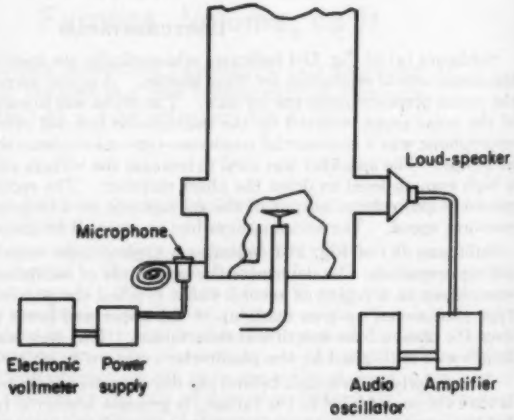


FIG. D-2. . . .SCHEMATIC SKETCH OF INSTRUMENTATION USED TO DETERMINE THE RESONANT FREQUENCY OF THE FURNACE

voltage output of the microphone was determined by an electronic voltmeter. The frequency of the sound source was varied until a maximum voltage output from the microphone was recorded. If none of the components of the instrumentation had a resonant frequency equal to that of the resonant frequency of the furnace, then that frequency which produced the maximum voltage output from the microphone was the determined resonant frequency of the furnace.

DISCUSSION

W. W. POPYK, Johnstown, Penna., (WRITTEN): The authors should be commended for an excellent report summarizing the results of research on combustion oscillations in single-port gas-fired heating units.

Of considerable interest is the fact that objectionable pulsation in this type of burner and heating unit originates with vortex shedding at the burner outlet coupled with a pressure fluctuation in the heating unit.

In the interest of possible methods of suppressing objectionable pulsation, could the authors comment just a little further, particularly in the respect of devices or shapes of burner outlet lips which may be considered. It is noted that reference is made to possibilities of such design considerations in the burner as well as some possibility in changing burner port diameter. The last item of burner port diameter appears to be contradictory to the authors' explanation of vortex shedding frequencies being dependent principally on average velocity of combustible gas flowing out of the burner tube and to a lesser extent, if any, on the burner port diameter. Possibly some undisclosed experimentation was conducted to suggest in the conclusion that vortex frequency can be altered by changing the burner port diameter.

It is also noted that changing the heating unit volume is referred to as a technique not normally of practical interest. Does this preclude any consideration of flueway baffles which may tend to break up the unit volume? Also, very often the unit volume beneath the burner level can be in excess of that required for combustion aeration. Could either of these 2 design areas be considered useful in suppressing oscillations?

AUTHOR'S CLOSURE, (Mr. Speich): In the paper several means were indicated by which the combustion oscillations in single-port burner-heating units could be suppressed. All of these techniques will be discussed in greater detail in a subsequent paper to be presented before the ASHRAE.

One suppression technique that has been found successful incorporates changes in the burner-port diameter. In particular, it was found that oscillations were successfully suppressed by either using an annular-type burner-port configuration, or by dividing the single-port into 3 ports without changing the net flow area for the combustible gases. This effect of burner-port diameter is not contradictory to the discussion accompanying Fig. 1 although an interesting point is involved. If the flow velocity is considered constant, no effect of burner-port diameter on frequency of the vortex-shedding is noted on Fig. 1. However, if the volume rate of flow of the combustible gases is considered constant, it can be shown that the frequency of vortex-shedding should vary inversely as the fourth power of the burner-port diameter.

Another burner port lip design which worked successfully incorporated a series of radially placed studs around the burner-port lip. These studs were placed so as to protrude into the flame. This design gave a marked decrease in the amplitude of oscillation and also caused the burner to exhibit excellent stability characteristics.

It should be realized that the combustion oscillations are basically an acoustical phenomenon and not directly connected with the combustion or heat transfer phenomena occurring within the heating unit. Therefore, changes made in the combustion system for the purpose of reducing the amplitude of the oscillations are made from an acoustic point of view. The compatibility of these changes with demands for good combustion and heat transfer must be considered separate from the acoustical demands, and possibly compromises made. Thus, such techniques as changing the

furnace volume can be considered as legitimate from the acoustical point of view for suppressing combustion oscillations. It is possible to change the internal furnace configuration by baffling, and similar means in such a way as to change the resonant frequency of the furnace. No extensive studies were conducted along this line. However, it is realized that this type of change may not be compatible with the heat transfer specifications of a particular burner-furnace design. Furthermore, it may be of value only when new furnace designs are being developed.

1658

No. 1658

LOW FREQUENCY COMBUSTION NOISE IN OIL-BURNING EQUIPMENT

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AN EXTENSIVE investigation concerning low frequency combustion noises in oil-burning equipment has recently been completed for a large manufacturer of domestic heating equipment which resulted in a considerable amount of information concerning the cause of this noise. This paper has been written to make available that information which should be of value to the entire heating industry. Most work done during this program was on commercially available domestic oil fired heating units with a fuel consumption rate of between 0.85 and 2 gal per hr. A large portion was at a rate of 1¼ gal per hr. Fig. 1 shows the furnace installations on which most of the experimental tests were made. Draft was provided by a large exhaust fan on the roof of the building with the airflow and static pressure in the draft system being controlled by a by-pass valve. This enabled a relatively constant draft to be maintained so that more accurate measurements could be made.

To present a concise description of the mechanisms and controls for low frequency combustion noise, this paper is prepared as a short analysis of the entire combustion system with the various facts concerning the mechanism and control of furnace noise being documented by experimental evidence where necessary. Much experimental documentation has been omitted to avoid excessive length.

The noise discussed here is the low frequency rumble present to some degree in all oil-burner installations and which makes itself objectionable by shaking doors, walls, and floors. This noise is usually audible in the basement or utility room and often in the living space. This combustion noise has a frequency of between 30 and 55 cps. The character of this noise is not that of a continuous tone; rather, it is more like a 40 cycle tone interrupted at random times in such a manner that the tone is heard approximately 30 percent of the lapsed time. It was not the primary purpose of this program to work on the more severe noise commonly called pulsa-

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tion. However, during the program the mechanism of pulsation was defined as a result of data obtained during the studies of rumble.

THE MECHANISM OF RUMBLE

Noise called rumble which is composed of 4 or 5 cycle bursts of a frequency near 40 cps is caused by pressure pulses in the fire box cavity exciting the fire box and flue system resonances. The pressure pulses which are due to rough burning produce an impulse type of sound which contains many frequencies. The burning noise itself is similar in nature to the noise produced by a random noise generator.

FIG. 1. . . .COMBUSTION NOISE RESEARCH INSTALLATION

In other words, there is no distinct single frequency characteristic to this noise: rather it contains almost all frequencies, at varying amplitudes, at varying times.

The fire box cavity and flue passages along with the oil burner itself make up a system analogous to a Helmholtz resonator. A system such as this has a single resonant frequency excited by the random pressure pulses caused by rough burning (see Fig. 2).

PULSATION

Pulsation is a noise similar to but not the same as rumble. It has the same exciting source, namely rough burning, and the same resonant system with one exception. In pulsation the fan and fan housing are involved to a greater degree than in rumble.

Pulsation takes place in the following manner. Random pressure pulses in the fire box increase the instantaneous fire box pressure so that it is higher than the

pressure in the blower housing (see Fig. 3). This momentary increase in pressure stops and then reverses the flow of air entering the fire box from the burner air tube. This causes a mixture of fuel and air too rich for combustion to be maintained in

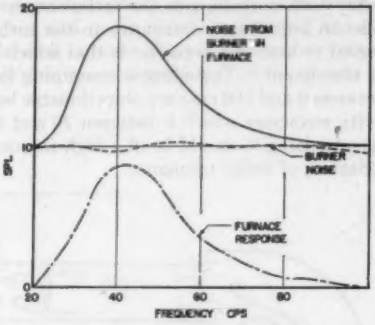

FIG. 2. . . .AMPLIFICATION OF LOW FREQUENCY COMBUSTION NOISE

the fire box. After a short period of time the pressure in the fire box is reduced by the escape of gases through the flue passages. As soon as the pressure in the fire box is reduced below the pressure in the fan housing, air rushes in and mixes with the unburned fuel. This causes a second pressure increase which again stops the

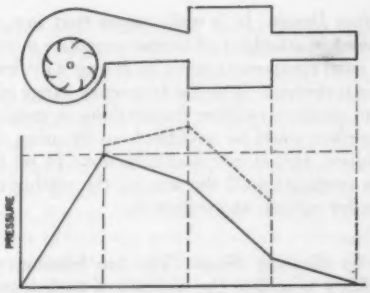

FIG. 3. . . .PULSATION MECHANISM

incoming air and causes the phenomenon to repeat itself until the conditions which produced pulsation have been eliminated. Pulsation frequency is determined by the decay rate of the pressure in the fire box which is determined by the resistance in the flue passages, the delivery characteristics of the blower, and the burning rate of the fuel.

THE NOISE SOURCE

Both pulsation and rumble are triggered by random fluctuation in the burning rate of the fuel fire and the air mixture. These fluctuations are caused by turbulent conditions in the air supply and mixing systems. In this system, the fan, fan housing, and drawer assembly each contribute to the turbulent air entering the fire box. The choke shape is also an important contributor to the turbulence.

The turbulence referred to here and hereafter is that which is undesirable from a rumble and pulsation standpoint. Turbulences containing large fluctuations at a low repetition rate (between 0 and 100 cps) are objectionable because they can easily excite the fire box cavity resonance which is between 20 and 60 cps. Since turbulence is required for good mixing it should be of a high frequency and small amplitude to reduce the excitation of boiler resonance.

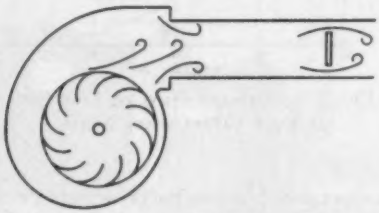

FIG. 4....FAN AND HOUSING SHAPE TURBULENCE

Turbulence Due to Fan Design: It is well known that fans with forward-curved blades, such as those used in standard oil burners produce a considerable amount of turbulence due to the swirl component given to the air as it leaves the fan blade, as shown in Fig. 4. This turbulence is in the frequency range of furnace cavity resonances, thus helping to produce random fluctuations in combustion in that range. These fluctuations in airflow could be smoothed out by using straightening vanes in the burner tube; however, this is not normally done in oil burner design. As a result, this turbulence propagates all the way to the mixing region and helps produce the exciting force for rumble and pulsation.

Turbulence Caused by Housing Shape: The fan housing normally used in oil burners is not streamlined to reduce the creation of turbulence. Sharp changes in cross-section cause eddies in the air stream, as shown in Fig. 4. These eddies are propagated through the burner tube and into the region of fuel and air mixing. This also produces part of the exciting force which causes rough burning and helps create rumble and pulsation.

Turbulence Produced by Drawer Assembly Shape: The drawer assembly of most modern oil burners is not designed to maintain a smooth continuous air stream. Some burners change the characteristics of the airflow by inserting flat plate baffles.

These baffles and other unstreamlined obstructions in the air passage cause turbulence which adds to noise level (see Fig. 4).

Fuel and Air Mixing: Fuel and air are mixed together in a manner determined by their patterns upon entering the combustion chamber from the burner tube. If either the fuel or air enters the mixing zone in an irregular or turbulent manner the mixing will be turbulent and uneven and the burning will have the same characteristics. Most modern oil burners direct the air toward the fuel spray by a choke which rapidly reduces the cross-sectional area of the burner tube. This reduction

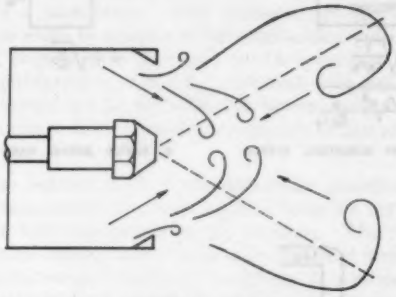

FIG. 5. . . .MIXING TURBULENCE

in area is sudden and produces large scale turbulence at the beginning of the mixing zone.

One of the major mixing problems in oil burner design is the configuration of the fuel pattern emanating from the nozzle. This fuel pattern can vary from a hollow to a solid cone of fuel. In a hollow cone of fuel there are two distinct mixing zones. The first is in the outside surface of the hollow cone where air leaving the choke meets the fuel. The second is the inside of the hollow cone which does not receive air directly from the air pattern created by the choke configuration. This inside zone receives its air either from the combustion chamber (causing air to move in a direction toward the fuel nozzle rather than in the normal direction of airflow), or the air gets to this inside zone of the hollow cone by being so directed from the choke that it ruptures the cone of fuel and enters near the fuel nozzle, as shown in Fig. 5. In this case the fuel pattern established by the nozzle is disrupted in an irregular manner, producing turbulence and rough burning. The rate of this rupture of the fuel cone determines the frequency components generated by the combustion. If this random interruption rate is in the vicinity of the resonant frequency of the fire box cavity, or is pulse-like in character having pulse shapes which generate harmonics near the resonant frequency of the fire box cavity, resonance will be excited, causing rumble. The random rate at which the air, not used in combustion at the outside surface of the cone, enters the center of the cone from the fire box, determines the frequency of the combustion ignition and extinction in the center combustion zone. The mechanism and locations of burning are similar, with a solid

fuel cone, since there is a similar outside surface and the inner zone, although less regular, contains a number of void spaces which are randomly supplied by air from the fire box area rather than from air supply at the choke.

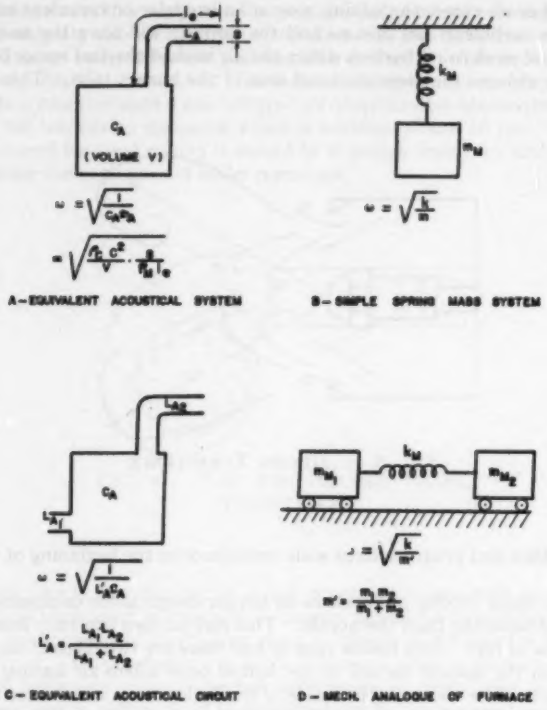
(VOLUME V)
A – EQUIVALENT ACOUSTICAL SYSTEM
B – SIMPLE SPRING MASS SYSTEM
C – EQUIVALENT ACOUSTICAL CIRCUIT
D – MECH. ANALOGUE OF FURNACE

FIG. 6. . . .ANALOGOUS ACOUSTICAL AND MECHANICAL SYSTEMS

This random mixing of fuel and air creates many areas in the burning zone which have extremely different fuel air mixtures and, consequently, different combustion rates. Random combustion is the basic exciting force which produces both rumble and pulsation.

Unstable Fuel Patterns: Although unstable fuel patterns were not a problem in any of the oil burners used during this investigation, it is pertinent to show how unstable fuel patterns, which might occur in experimental systems, would help generate combustion noise. There are 2 primary oscillatory systems in the oil supply. First is the positive displacement pump which produces pulsation in fuel

flow at a frequency equal to the pump revolution rate times the number of pump lobes. These pulsations in fuel flow were seen in high speed motion pictures of the fuel stream discharging from the nozzle. However, their magnitude was small and their frequency considerably above the range which might produce rumble. The second source of pulsation in the fuel system is the pressure regulator which produces changes in fuel pressure at a rate determined by the mass of the valve and the spring rate of the valve actuating mechanism when excited by a random pump fluctuation. Pulsations caused by the pressure regulator were not observed during this investigation; however, these might be present in some fuel burning systems.

Oscillatory Burning Due to Spark Ignition: Modern oil burners are ignited by a 120 cps spark from a transformer which produces 120 sparks per sec due to 60 cycle current. This tends to produce a fuel combustion pulse rate of 120 cps. If the fire box cavity happened to be tuned to this frequency there would be considerable noise amplification at 120 cps. Although this was not the case in the test unit, this fact should not be overlooked in future designs expected to result in smaller units which might have a resonant frequency in this range.

Summary of Noise Sources: Each of the turbulence generators mentioned in the preceding sections helps create the basic exciting force for both rumble and pulsation. All are important contributors to the problem. The mixing process itself, especially in the central mixing zone, seems to be the most important. In order to produce a quiet oil burner it is necessary to reduce to a minimum the creation of low frequency turbulences and to rely fully on high frequency turbulence to provide the mixing necessary for efficient combustion. In the Appendix there is a list of suggested methods for reducing low frequency turbulences.

EFFECT OF FIRE BOX AND FLUES ON RUMBLE

Previous sections have mentioned the parameters involved in creating the rumble exciting force. Since all of the mechanisms mentioned produce random combustion noise in the burning zone which is inside of the unit, the effect of this internal volume must be considered in describing the rumble noise as it is heard in the adjacent space. The unit consists of a fire box, flue system and draft openings. To this is attached the oil burner, which constitutes the second major opening in the system. From an acoustic standpoint, this system is analogous to a Helmholtz resonator with 2 openings. The fire box is the cavity or acoustic compliance. The two masses or tubes of the Helmholtz resonator are the burner tube and flue system. For the purpose of rumble, the system may be considered terminated at the draft opening or barometric damper. This point may be considered the termination because changes in length and diameter of the stack beyond the barometric damper do not greatly influence the resonant frequency or the amplification of rumble.

Effect of Internal Configuration on Frequency Rumble: Fig. 6 shows a diagrammatic sketch of a Helmholtz resonator, along with the simplified equation which determines its resonant frequency.

$$\omega = \sqrt{\rho_0 C^2 / V \cdot S / \rho_0 l_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

ρ_b = density in the fire box.
ρ_M = density in the flue.
V = volume of the fire box.
S = cross-sectional area of the flue.
C = velocity of sound.
l_e = effective length of the flue.

From the equation for the resonant frequency of a Helmholtz resonator it can be seen that an increase in the size of the combustion cavity or fire box decreases the resonant frequency of the system. It can also be seen that an increase of the length of the flue passes decreases the resonant frequency. The converse of both of these

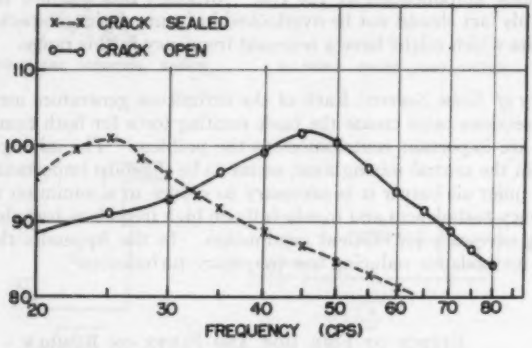

FIG. 7. . . .THE EFFECT OF CRACKS ON FURNACE RESONANT FREQUENCY

statements is equally obvious. The acoustic mass of the system is actually a combination of 2 individual masses operating in parallel. The way they are added together to form the effective mass is shown in the equation below Fig. 6-C. It can be seen from this equation that it is possible to have either the flue passages or the burner tube constitute the more important portion of the effective mass. When either is more important a change in it naturally has greater effect on the resonant frequency of the system. In the case of the test unit the flue passages are a little more important than the burner tube in determining the resonant frequency of the unit. Calculations and measurements of the resonant frequency of the test unit show that when the unit is cold and well sealed it has a resonant frequency of between 20 and 25 cps. Density of the gases inside the unit affects the resonant frequency. While burning with an average temperature distribution in the system the resonant frequency is approximately 40 cps instead of 20 to 25 cps. This is due to the change in density caused by the increased temperature of the gases in the flue passage. Density of the fire box gases is unimportant since density is so used twice in the equation for the compliance of the fire box that the effect of tem-

perature is cancelled out. With regard to the masses involved, it is only in a sealed unit that the flue passes and the burner tube are the primary ones involved. As soon as cracks are opened the total mass of the system changes radically. Measurements of the natural frequency of the unit with several cracks of less than half of 1 sq in. in cross-sectional area raised the resonant frequency of the cold unit to approximately 45 cps, as shown in Fig. 7. Because cracks have such a large effect on the resonant frequency of the unit, it is not possible to measure the resonant frequency of a particular unit with any degree of accuracy unless attention is paid to sealing all cracks which might change the tuning. All of the factors just mentioned determine the frequency of the rumble. The combustion noise, if it were outside the unit, would have a relatively broad band spectrum in the low frequency range. However, when combustion takes place inside, the unit amplifies the noise at its resonant frequency. Therefore, when one listens in the basement or utility room, the rumble tends to have definite frequency characteristics determined by the unit configuration.

Effect of Unit Configuration on Rumble Amplification: Any resonant system, such as a heating unit, has a sound-pressure amplification factor, Q. The Q of the system might be roughly described as the number of times a particular input signal is amplified inside the resonant system. In the test unit it was found that this amplification factor, or Q, was between 1½ and 4. This would represent a sound-level amplification of between 3 and 12 db, 3 db being for a Q of 1.5 and 12 db being for a Q of 4. One reason for the variation in noise from so-called identical units is the number of unsealed small openings or leaks. Leaks, beside helping determine the resonant frequency of the system, also help determine the amount of damping the system has.

Leaks produce damping by two means. First, there are viscous losses in the narrow passages. The loss produced by viscosity of the fluid is proportional to the particle velocity in these small openings. The resistance of a small opening can be calculated from Equation 2.

$$R_a = 8Nl/\pi a^4 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

R_a = acoustic resistance.
N = viscosity coefficient.
l = length of the opening in the direction of flow, and a is the equivalent radius of the opening.

Leaks also produce resistance or damping due to an airflow phenomenon. Resistance created in the leaks due to airflow is proportional to the linear velocity of d-c airflow through the opening divided by the cross-sectional area of the opening. Equation 3 for this particular resistance was unknown prior to the work on this program. Work carried out at the Foundation under its own sponsorship during the course of this program has established that the acoustic resistance due to d-c airflow can be calculated by the equation

$$R_a = 2.12 \times 10^{-3} V_{DC}/S \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

V_{DC} = d-c velocity in centimeters per second.
S = cross-sectional area of the opening or hole in square centimeters.

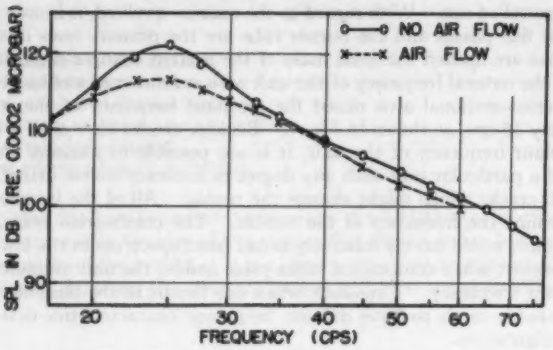

FIG. 8. . . .THE EFFECT OF AIR FLOW ON FURNACE AMPLIFICATION

In a unit the d-c airflow through leaks is created by the pressure difference between the surrounding room and the inside of the fire box. It makes no difference whether this pressure difference is positive or negative since either will cause air to flow through leaks into the unit in an attempt to reduce the pressure difference. Normally there is a negative overfire draft causing air to flow from the room into the fire box. The effect of this d-c airflow on the Q of the furnace is shown in Fig. 8. This figure correlates the change in sound pressure in the fire box with a change in overfire draft. Note that a higher draft produces a response curve which indicates lower noise amplification at the resonant frequency of the unit.

D-c airflow through the unit changes the damping in the system by creating acoustic resistance at locations other than leaks. The airflow required for combustion flows through the burner tube and the flue passes. This creates acoustic re-

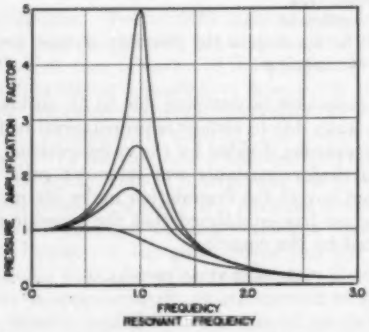

FIG. 9. . . .TYPICAL RESPONSE CURVES OF A SIMPLE RESONANT SYSTEM

sistance in both of these locations. The resistance created in the burner tube is more important than the resistance created in the flue passes because of the higher air velocity in the burner tube. It was mentioned in the section concerning the effect of the burner tube on rumble and pulsation that a small cross-sectional area is desirable in the burner tube. A second reason for the importance of this small cross-sectional area is to increase the d-c velocity at that location so that increased acoustical resistance may be added to the resonant system. This is important from both a rumble and pulsation standpoint. It is important from a rumble standpoint because of the reduced Q of the entire system. It is more important from a pulsation standpoint, however, because in addition to reducing the Q of the system the resistance in the throat of the burner tube damps and, thereby, reduces the amplitude of any reverse flow toward the fan. The total resistance in the burner tube is the sum of the a-c resistance due to d-c airflow and the a-c resistance due to

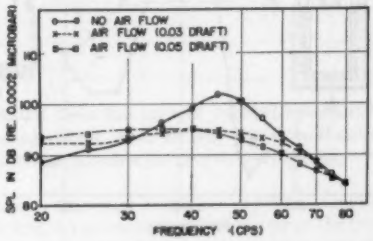

FIG. 10. . . . TYPICAL FURNACE RESPONSE CURVES

viscous losses in the throat of the tube. When these resistances are increased by reducing the throat cross-section, the maximum amplitude of pulsation is reduced because of the additional damping.

The effect of damping on any resonant system can be visualized by inspecting a curve such as Fig. 9. This curve shows the change in amplification of a simple series resonant system with a change in damping or resistance. A heating unit acoustical system is not as simple as the curve indicates; however, the similarity is close enough so that a similar curve could be plotted for the exact unit being studied (see Fig. 10).

In general, most units today are similar from the standpoint of size and noise amplification. This can best be illustrated by Fig. 11, which shows a sketch of the cross-section of 2 extremely different designs along with curves of the noise output of each unit using the same oil burner and smoke stack. This comparison indicates how little can actually be done by changing internal configurations of the unit to reduce noise.

Radiation of Combustion Noise: There are several locations at which rumble and pulsation are radiated from the unit to the surrounding room. The first and usually most important is the opening at the barometric damper. In most installations this can be verified simply by listening to the rumble in the room with the barometric

damper open and closed. There are, however, several other important radiation points. These include cracks, the smoke pipe wall, the air inlet on the burner, and any auxiliary ports which might be placed in the fire box to control pulsations and starting noises. The radiation resistance of sound from an opening which is small

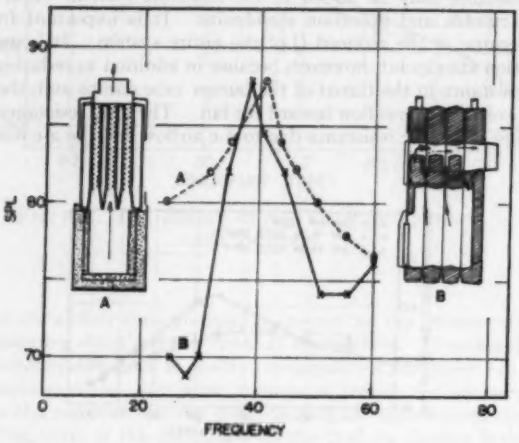

FIG. 11....EFFECT OF FURNACE CONFIGURATION ON LOW FREQUENCY NOISE

compared to the wavelength of the sound being radiated may be calculated by Equation 4.

$$R_{RAD} = S^2\rho\omega^2/2\pi c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where

S = cross-sectional area.

ρ = density of the gas.

ω = frequency.

c = speed of sound.

The amount of sound radiated at any opening is proportional to the radiation resistance at that point. A series of measurements was made to determine the effect of damper size on sound level radiated to the room. As expected, these indicated that a large opening radiates more noise than a small opening. Measurements on the test unit indicate the difference in the sound power radiated from various smoke pipe configurations as shown in Table 1. These results generally agree with the theoretical equations for sound radiation and should be considered in the design of any new heating unit.

Noise Considerations in Heating Unit Design: In order to insure a minimum of rumble complaints the design of units should include features which will give the most desirable results from an acoustic standpoint. There are 2 specific changes in rumble which will reduce complaints. First, the basic rumble frequency should be increased so that it will be out of the range of resonant frequency of doors, floors, and walls. It is felt that a resonant frequency in the neighborhood of 100 cps would

TABLE 1. . . . EFFECT OF DAMPER AND SMOKE PIPE SIZE

SMOKE PIPE DIAM, INCHES	DAMPER DIAM, INCHES	SOUND-PRESSURE LEVEL AT DAMPER, DB	RELATIVE POWER LEVEL IN FURNACE ROOM, DB
6	6	74	123
8	6	77	129
8	9	75	131

be much less objectionable than the present typical resonant frequency of 40 cps. The second major consideration is to reduce the amplification and radiation of combustion noise as much as possible by proper unit design.

1. *The Increase of Furnace Resonant Frequency* can be accomplished in 2 basic ways. The first is to reduce the length and increase the diameter of the flue passages as much as possible. The second is to reduce the fire box volume; this type of change will shift the furnace resonance away from that of doors and walls of the building being heated.

2. *Reduction of Noise Amplification* in the unit tested, which had a low Q, was not feasible by changing the design of the unit. It was possible, however, to reduce the susceptibility of the unit to pulsation by changing basic design. This can be accomplished by increasing the cross-sectional area of the flue passages so that a specific pressure pulse in the fire box will be limited in amplitude because of more rapid release of the confined gases.

3. *Reduction of Noise Radiation* is possible by reducing the size of all openings between the fire box and the room. Since the barometric damper is generally the largest opening from which the noise emanates, it is desirable to reduce its size as much as possible. This does not mean that the smoke pipe size should be reduced but only that the size of the damper opening itself should be reduced. If a certain amount of acoustic energy is leaving the unit, this energy will be distributed into the various branches of the smoke pipe, depending on the size of these branches. A smaller amount of the acoustic energy will flow through the smaller branch. Therefore, it is desirable to have the barometric damper opening small compared to the size of the smoke pipe. This will allow most of the acoustic energy to go up the stack rather than into the room. This analysis is based on the assumption that there is very little radiation into the room from the smoke pipe walls. Since this assumption is not strictly valid, and since some energy gets to the room from the smoke pipe walls, it is desirable to have the length of smoke pipe in the furnace room as short as possible. The smoke pipe is thin walled and transmits low frequency much more effectively than brick or masonry chimney.

Because of the thin walled construction of most smoke pipes, the acoustic end of the resonant system is not necessarily at the barometric damper. The thin pipe walls

permit sound to be radiated and, therefore, the termination of the system takes place close to the outlet of the unit whether the barometric damper is in that location or elsewhere. Fig. 12 shows the effect of moving the damper away from the outlet in the particular configuration investigated. It will be noted from the curve that the resonant frequency of the fire box has not changed, although the radiation at the lowest frequency resonance has been reduced by moving the barometric damper away from the back of the unit.

EFFECT OF THE ROOM ON RUMBLE AND PULSATION

In almost all installations it is noticed that rumble can be heard much more distinctly in the corners of the room rather than in the center of the room. This is

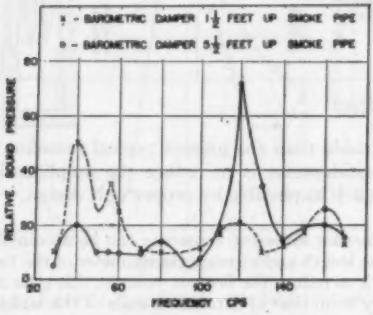

FIG. 12. . . .EFFECT OF BAROMETRIC DAMPER LOCATION ON FREQUENCY SPECTRUM OF RUMBLE

due to the increase of sound pressure near a reflecting surface. This amplification or pressure increase has been described fully in the literature. Waterhouse† gives a good summary of the phenomena involved. It is sufficient to say that, close to a single wall, the sound pressure increases 3 db. Close to a corner the sound level increases 6 db and increases 9 db close to a tri-axial corner. The word close in this case refers to a distance that is small compared to a wavelength of the sound frequency involved. The phenomena mentioned here and described in the references apply to the sound pressure with respect to a source located near the center of a hard walled room. It has also been shown that a source which is placed near a wall corner or tri-axial corner produces 3, 6, or 9 db more sound than would the same source in the center of the room. This is also described by Waterhouse‡. It is not possible to reduce the sound preasure amplification near a wall or corner from a

† R. V. Waterhouse: On standard methods of measurement in architectural acoustics (Acoustical Society of America, Vol. 29, 1957, p. 544).

‡ R. V. Waterhouse: Out-put of a sound source in a reverberation chamber and other reflecting environments (Acoustical Society of America, Vol. 30, 1958, p. 4).

source in the center of the room. However, it is possible to reduce the amplification of sound which would take place if the source were placed in the corner of the room. It is, therefore, suggested that where possible heating equipment be installed as far as possible from any wall in order to reduce furnace rumble between 3 and 9 db from what it would be with the source near a wall or corner. The best location would be approximately ⅛ to ½ wavelengths from the nearest wall. For a resonant frequency of 40 cps this would be a distance of approximately 7 to 12 ft. It is realized this would often be impractical, but in certain situations may be helpful.

CONSIDERATIONS FOR THE REDUCTION OF PULSATION

To prevent pulsation, the head produced by the fan must be greater than the head loss in the blast tube plus the head loss in the flues plus the pressure of any random pulse created in the fire box by rough combustion. Once the sum of these 3 pressures exceeds the pressure delivered by the burner fan continuous pulsation will take place. Therefore, it is desirable to have a fan which delivers a substantial pressure, perhaps as much as 2 to 3 in. of water. It is also desirable to have as small a head loss as possible in the flues. Having a small flow resistance at this location helps prevent excessive pressures in the fire box as a result of random fluctuations in burning. The amplitude of the random burning fluctuations should be kept to a minimum in a manner similar to that described in the sections concerning the rumble exciting force.

A high pressure drop should be maintained in the burner tube even though this requires a larger fan to deliver sufficient air to the mixing region. A high head loss at this location does 2 things: first of all, it reduces the effect on the fan of any pressure pulsation in the cavity; it also requires that there be a high velocity in the throat to the burner tube. This high velocity creates acoustic resistance in the burner tube and, thus, absorbs some of the energy which might try to build up the pressure at the fan discharge with consequent reduction in fan delivery. The high velocity in the throat of the burner tube results in a high velocity head which opposes any reversal in airflow. This high velocity head is just as efficient as high static head in reducing the effect of pulsations in the fire box on the mass flow of air into the mixing region provided the throat area is small. The ideal but impractical solution to the problem would be to have a 100 psi air supply throttled through a very small orifice in the burner tube. In this case, a pressure fluctuation in the fire box of perhaps $\frac{1}{10}$ lb per sq in. would only change the pressure drop across the orifice $\frac{1}{10}$ of 1 percent. This would have a negligible effect on the mass flow through the orifice and would insure that the air delivered to the mixing zone was constant within the $\frac{1}{10}$ of 1 percent. In order to cause a system like this to pulsate it would be necessary to restrict the flue passages enough so that the static pressure in the fire box was increased to 99.9 lb per sq in. To reduce the amplitude of any pulsation which might take place, it is desirable that the fan have a steep pressure-volume curve. This type of curve can be obtained with a fully backward curved fan. The steep pressure-volume curve reduces pulsation amplitude by resisting any tendency for airflow to decrease. A sudden reduction in volume flow increases the fan delivery pressure very sharply and, thus, creates more energy to force the air through the throat of the blast tube. A fan with a flat pressure flow curve does not give this added energy to the system.

ACKNOWLEDGMENTS

During the course of this program almost everyone in the acoustics group made a substantial contribution. The work done was directed by Drs. H. C. Hardy and R. W. Benson, and supervised by the author. The analyses and test work were performed by numerous members of the group, with major roles being played by W. E. Lawrie, E. G. Grimsal, G. L. Bonvallet, and C. S. Caccavari. The advice and council of F. G. Tyzzer was very valuable during the course of the entire program. The staff of the Foundation is indebted to M. W. McRae and John G. Pecis, of Crane Co., for their guidance of this program and, also, for the procurement of numerous pieces of apparatus.

APPENDIX

SUGGESTED METHODS FOR REDUCING LOW FREQUENCY TURBULENCES

Fan Turbulence: It is suggested that, where possible, oil burners with fully backward curved fans be used. Besides producing a less turbulent air stream the fully backward curved fan has a more desirable pressure flow characteristic. Its characteristic is compared with that of a forward curved fan in Fig. A-1. As can be seen from the figure, the forward curved fan characteristic has a dip in the middle of its operating range. Whenever the airflow from the fan is such that it is operating on this reverse slope, fan operation is unstable and pulsations in airflow are likely to take place. On the other hand, there is no reverse slope in the backward fan characteristic and, therefore, pulsation is not likely.

Fan Housing Shape: All changes in cross-sectional area in the fan housing should be gradual so that there are no sharp discontinuities to produce eddies. The fan housing should also fit the fan blade on the sides without leaving an excessively large by-pass area for the discharge air to re-circulate back to the inlet.

Drawer Assembly: The drawer assembly and blast tube must be streamlined so that no low frequency turbulence can be generated due to the abrupt discontinuity in the air stream. It is also desirable that the airflow be regulated in the drawer assembly by choking rather than at the fan inlet. Regulation of flow in the drawer assembly necessitates the smallest possible cross-sectional area in this location; the smaller the cross-sectional area the greater the damping of any turbulence which was created in the fan or fan housing.

Improved Fuel and Air Mixing: Once the fan and blast tube have been designed so that very little turbulence is created in them it can be assumed that a quiescent stream of air is present for combustion. If it is also assumed that the fuel pump and pressure regulator have been designed properly a smooth cone of fuel will also be present. It is next necessary to mix these two together. The mixing must be uniform and, therefore, necessitates uniform turbulence throughout the mixing regions. Since turbulence is required for proper mixing, and since turbulence also produces noise, it is essential that the frequency of the turbulence used for mixing be out of the frequency range where it could produce combustion noise to excite the resonance of the unit.

Effect of Spark on Smooth Burning: Ignition of the combustible mixture by continuous spark is not necessarily desirable acoustically but is a safeguard to permit immediate

re-ignition after a flame failure. The normal spark ignition provided in oil burners produces 120 cycle intermittent combustion in the area close to the electrodes. This is not a serious problem with conventional units today. However, it may be a problem when the size of the units has been reduced to a point where the resonant frequency of the fire box and flue passage is in the neighborhood of 120 cps.

CALCULATING RESONANT FREQUENCIES OF SIMPLE TUNED SYSTEMS

Resonances in acoustic systems can occur just as in mechanical or electrical circuits, and the equations showing the relationships between the forces, or pressures, and

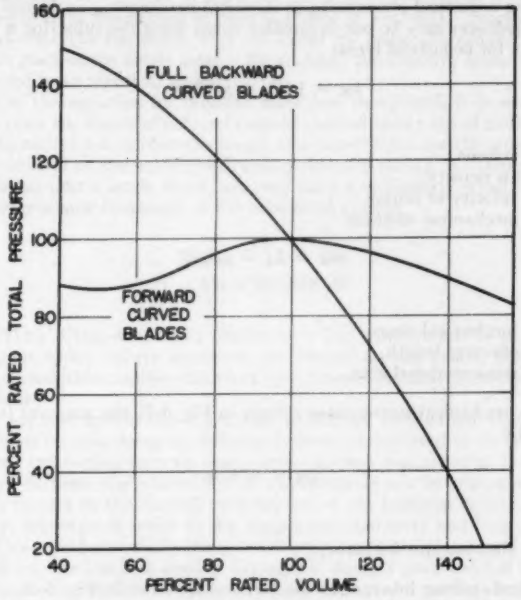

FIG. A-1. . . .TYPICAL FAN WHEEL CHARACTERISTICS

velocities, either particle or volume, are identical in form to the equations which express the ratio of *e.g.*, voltage to current in an electrical circuit.

Resonance in any system is characterized by 2 conditions; one, the force and velocity (voltage and current) are in phase and, two, the velocity for a given force (current for a given voltage) is either a maximum (resonance) or a minimum (anti-resonance).

If the impedance is composed of only dissipative elements, no resonance can exist. Only when the impedance elements are reactive can the resonance take place. Reactive elements are frequency sensitive, create a phase shift between the force and velocity, and can store energy but not dissipate it.

To predict resonances and compute impedances, pressures, or velocities in acoustics, analogies are frequently used. Although both mechanical and electrical analogies are

used, the electrical analogy is generally preferred since electrical circuit theory is slightly more advanced than is mechanical theory. However, either may be used since the main purpose of an analog is to put an unknown problem into familiar terms.

In acoustics, 3 distinct impedances are used; the analogous acoustic impedance, the specific impedance, and the mechanical impedance. The *first* is used with lumped constant systems, *i.e.*, systems where the dimensions are small compared to a wavelength. The *second* is used when the dimensions are comparable to a wavelength, and the *third* is used when working with both acoustical and mechanical systems. When considering the acoustical properties of residential boilers, it is generally not necessary to consider distributed systems. Normally, the dimensions are small compared to the wavelengths of frequencies of interest so that analogous impedances are applicable. Mechanical impedances should be used when considering radiation of sound.

By comparing acoustical phenomena to electrical phenomena, analogous reactive and dissipative impedances may be put in familiar terms using the subscript M for mechanical terms and A for acoustical terms

$$k_M = 1/C_A = \rho_V C^2/V \quad \text{(A-1)}$$

where

V = volume.
ρ = air density.
C = velocity of sound.
k_M = mechanical stiffness.

$$m_M = L_A = \rho_L l_e/S \quad \text{(A-2)}$$

where

m_M = mechanical mass.
l_e = effective length.
S = cross-sectional area.

Considering a mechanical spring-mass system in Fig. 6-B, the resonant frequency may be computed by

$$\omega_R = \sqrt{k/m} \quad \text{(A-3)}$$

where

ω_R = the resonant frequency.

The acoustical system, inherent in the furnace, is shown in Fig. 6-A. The resonant frequency is found by an analogous equation, with k_M and m_M replaced by the analogous acoustical quantities just given, so that

$$\omega_R = \sqrt{\left(\frac{\rho_V C^2}{V}\right)\left(\frac{S}{\rho_L l_e}\right)} \quad \text{(A-4)}$$

where

ρ_V = the density in fire box.
ρ_L = the density in the flue.

If the flue and the fire box are at the same temperature, then the densities ρ_V and ρ_L are equal and the frequency reduced to

$$\omega_R = C\sqrt{\frac{S}{Vl_e}} \quad \text{(A-5)}$$

If the temperatures are different, the factor C^2 in Equation A-4 may be replaced by $\gamma P/\rho v$, so that

$$\omega_R = \sqrt{\left(\frac{\gamma P}{V}\right)\left(\frac{S}{\rho l_e}\right)} \quad \text{(A-6)}$$

where

γ = ratio of specific heats.
P = average pressure.

If, in addition to the mass of the flue, additional masses produced by the burner or extraneous openings are present in the unit, the acoustical diagram is as shown in Fig. 6-C. The equivalent mechanical system is shown in Fig. 6-D. Standard mechanical theory shows that the frequency may be found by using the reduced mass, $m_1 m_2/(m_1 + m_2)$, in place of the single mass. By analogy, the effective mass in the acoustical system is found in an identical manner.

Referring to the equation for reduced mass just mentioned, it is seen that if m_1 is much larger than m_2, then the reduced mass is approximately equal to the smaller mass m_2. From Equation A-2, neither the length of a constriction nor the cross-sectional area has as much control on the equivalent mass as does the ratio l_e/S. Examination of the expression shows that a small, short hole may have a sufficiently small equivalent mass to control the resonant frequency of the acoustical system.

DISCUSSION

W. M. MYLER, Columbus, Ohio (WRITTEN): The authors are to be complimented upon this paper which clearly separates the several factors which, working together, produce the undesirable rumble which has been present in many domestic oil-fired heating installations.

It is very easy now to see why it has been so difficult to eliminate this rumble by cut and try methods because so many different factors are involved in its creation.

It is further interesting to note that certain actions suggested as being the proper thing to do to decrease the occurrence of rumble would not be considered good design practice with respect to the heating performance of the boiler or furnace.

Specifically, reference is made to the suggestion of shorter and larger diameter flue passages. This would materially increase the problem of obtaining good thermal transfer. In addition, the use of a smaller barometric damper than the full size of the flue pipe, could make it difficult to hold the stack draft down to the desired maximum, particularly on a tall chimney.

Apparently a compromise must be reached between a design which will hold rumble characteristics within acceptable limits while at the same time permitting reasonable thermal efficiencies. It is a challenging problem.

E. M. MITTENDORFF, Charlottesville, Va. (WRITTEN): I would like to have the authors clarify the section on *Pulsation*, particularly the effect on fan discharge.

Under the heading, *The Noise Source*, I would like to ask just what is specifically meant by *burning rate of the fuel fire and the air mixture*. Do the authors mean what is more commonly known as fuel-air mixture?

What is indicated by the dotted curve in Fig. 3?

The paragraph devoted to the *Oscillatory Burning Due to Spark Ignition* gives the impression that the spark maintains the fire. I would say that, with a hot fire box cavity, the fire would continue without a spark and wonder if the authors would explain why they feel that cyclic sparking would contribute to continuing noise.

After Equations 2 and 3, the authors talk about d-c and a-c airflow. I am not sure that I understand fully just what is meant by that.

The same applies to the symbol SPL in Figs. 7 and 8.

W. W. POPYK, Johnstown, Pa. (WRITTEN): The paper, represents a very complete scientific investigation of objectionable combustion noises originating with oil firing in contemporary heating equipment. The cause of this noise further attributed to certain adverse design conditions found in the oil burner, combustion chamber and flue passages of a heating appliance should prove to be of immense value.

There is no doubt but that many heating-equipment manufacturers concerned with the problem of combustion noises have made similar findings by trial and error methods. It is doubtful, however, whether any individual manufacturer has ever attempted to correlate as complete a study beginning with the firing device and carrying through the heating appliance and the chimney connection. All of these factors in a heating unit assembly have an apparent relationship to one another as indicated in the paper.

It would appear as though the findings of this paper with some further amplification could be used as an effective guide in design and development by the manufacturers of the various heating equipment components. Some of the conclusions could be used in the field by installers and servicemen in minimizing and overcoming combustion noises.

It is noted however, that no mention is made of modifying an oil burner, heating appliance or combustion chamber in accordance with some of the deductions and findings of this report. It would be of considerable interest if the authors could comment in this respect and also whether any one factor or combination of factors may be more conducive to generating objectionable combustion noises.

It is recognized that very often several factors can be significantly involved in contributing to the noise and therefore, more specific comments by the authors would be helpful in appraising design and installation conditions for instituting corrective measures in overcoming combustion noises.

C. F. SPEICH*, A. A. PUTNAM†, D. W. LOCKLIN, Columbus, Ohio (WRITTEN): The paper by Messrs. Sanders and Lawrie was of great interest to us, especially since the authors' work parallels a portion of an ASHAE-sponsored program just completed at Battelle Memorial Institute concerning the study of oscillations and pulsation in residential heating equipment, both oil- and gas-fired. Through M. W. McRae, who was both the sponsor representative to the authors' project and who is currently chairman of the Technical Advisory Committee on Combustion that sponsored the ASHAE program, the discussers had the opportunity on several occasions to confer with Mr. Sanders and his staff regarding this problem of combustion-driven oscillations in oil-fired heating equipment. Also, through Mr. McRae, the authors have been able to keep abreast of the Battelle work through our periodic reports to the TAC. Although the project at Armour Research Foundation was directed to a specific class of units, it is felt that the exchange of information was of benefit to the ASHAE-sponsored project and the discussers wish to acknowledge this cooperation.

Basically, the authors show the importance of the resonant frequency and the large effect of air-leakage cracks in the heating unit on the resulting amplitude of the pulsation. They have credited turbulence in the air stream as being the primary cause of rumble‡ and under adverse conditions, pulsation‡.

The discussers' findings have confirmed the dominance of the resonant frequency of the furnace-burner unit. Both their calculations and experimental results have shown that this resonant frequency is quite sensitive to changes in the internal configuration of the burner. During part of the studies, a manifolding chamber was placed over the

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† Assistant Division Chief, Battelle Memorial Institute.

‡ Believed to be equivalent to the terms "low- and high-amplitude oscillations" described in ASHAE RESEARCH REPORT No. 1596—Pulsations in Residential Heating Equipment—Preliminary Results, by A. A. Putnam and W. R. Dennis (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 153).

normal air inlet and this, also, was found to affect the acoustic behavior of the furnace-burner unit.

The authors propose as the driving force for the *rumble* the presence of large scale turbulence which is produced by abrupt changes in cross-sectional area, such as in the fan housing and interior of the blast tube. We would like to suggest that much of this turbulence may be considered as a form of a periodic vortex shedding; thus, the normal frequencies would be a function of an average flow velocity and the internal configuration of the burner. Work by Garber[¤] has shown that such vortex shedding can often be *locked in* to an acoustic frequency of a unit. Our investigations on single-port, gas-burner furnace units appear to confirm this effect. Therefore, increases in the average velocity, such as might be caused by increasing the pressure drop across the blast-tube assembly, would not necessarily be a solution to the problem of eliminating this potential driving force. Some means would be needed to break up these vortexes into small-scale turbulence. As pointed out by the authors, small-scale turbulence is necessary to good mixing. For instance, passing the turbulent air through a grid might be one method of producing this small-scale turbulence.

The authors suggest that backward-pitched fans be employed in oil burners as a means of producing the desired pressure-flow capacity relation for the volume flow rates normally used in these burners. For the fan speeds and sizes normally used and the loose scrolling employed, our experience has been that bladepitch appears to have only a small effect on the noise amplitude and that this effect occurs at abnormally rich mixture ratios.

A small blast-tube orifice, of the type suggested by the authors for use with an upstream pressure of 100 psi, would act as a critical flow orifice when operating at any pressure ratio across the orifice greater than 2. For this case, a pressure pulse could not be transmitted upstream through the orifice. It is agreed that this general type of solution would not be practical, except possibly where a sufficient supply of high pressure air is already available. Furthermore, if the volume between orifice inlet and blast-tube exit were too large, the discussers' work on such a system indicates that pulsations could still occur.

The authors suggest that a smaller barometric damper opening will reduce noise radiation. In practice, the damper opening area is generally not amenable to choice by the designer since the barometric damper blade will seek the opening required to control draft to the desired level, unless the opening available with a full-open damper is inadequate to control the draft for the condition involved. In the latter case a larger barometric damper is needed.

We wish to inquire about several items in the paper which the authors may wish to clarify further. In Fig. 7, does the same reference level for sound pressure apply for both curves? Data for Figs. 7, 8, and 10 appear to have been obtained with a cold unit; we wonder whether data for Figs. 11 and 12 were obtained with combustion. It is suggested that a reference for the use of Equation 2 be given since this relation is not easily found in standard texts. This would also indicate the use of Equation 3. Without such references, it would be difficult for the designer to apply these relations in calculating flow resistances for burner units; an actual example might therefore be advisable.

In general, the findings in the ASHAE-sponsored program are in agreement with most of the results of the authors investigation, as will be revealed in a paper now in preparation. However, the terminology and interpretation are somewhat different on some minor points. Such plurality of viewpoints is felt to be helpful in stimulating study toward the understanding of the general nature of the pulsation problem.

The results of both of these studies emphasize the dilemma of the designer, in that the means for suppressing objectionable noise is often contrary to other important considerations such as smoke-free combustion, heat transfer, compactness, convenience in location, and cost. For example, the increase in flue-passage area suggested by the authors may

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¤ Donald I. Garber, The effect of external sound on the vortex-shedding from cylinders, *Journal of the Aeronautical Sciences*, Vol. 25, No. 4, April, 1958, p. 275.

be expected to reduce heat-transfer rates and increase the physical size of the unit. Each particular case involves, therefore, a compromise of various factors.

R. W. SAGE, Linden, N. J.: I would first like to take this opportunity to compliment the staff at Armour Research Foundation on a job well done. It is felt that they have made a valuable contribution to oil heating engineering. Particularly impressive is the great detail of the work and the large amount of effort that went into it.

We at our laboratories have also been working on this problem of noise, and particularly on pulsation. I would like to spend a few minutes comparing the authors' results with those obtained in our work.

In general this discusser agrees with the authors' conclusions. The authors' theories on the basic mechanism of pulsation—why these pulsations occur and how they are perpetuated—coincide exactly with ours. We have also taken high speed motion pictures of a pulsing oil flame which clearly support these theories. In these pictures one can see the flame flickering off and on at a frequency of about 30 cps. At the peak of the pressure surge when the flame is dead for lack of air, one can see the flow of gas back into the blast tube.

The moving pictures also reveal the turbulence and instability of even an ordinary nonpulsing flame. We concluded as did the authors that this unstable combustion is the basic exciting force for both rumble and pulsation.

Incidentally, in our studies it was considered that this rumble is an incipient pulsation and it was found, that if one just pushes rumble a little bit more it becomes pulsation, and if one cuts the pulsation just a little the result is a return to the rumble region.

We also agreed that probably the most important cause of this low frequency turbulence is the mixing process itself where the air and oil are brought together. A number of the expedients suggested by the authors to reduce the turbulence of the air in the blast tube were tried, but with little success. The problem seems to be more in the mixing in the combustion chamber itself.

The problem of obtaining good mixing without excessive noise is a tough one. The authors suggest minimizing all low frequency turbulences and relying completely on high frequency turbulence to provide the mixing necessary for efficient combustion. This will, no doubt, reduce the low frequency noise, or rumble as they call it, and should keep the doors and windows from shaking. But it will do this at the expense of increasing the higher frequency noises.

This discusser's experience indicates that the noise spectrum for an average oil burner is roughly parallel to the so-called equal annoyance curve. So that, if one increases the high frequency noise much there is trouble again.

It is believed to boil down to the simply stated—though not easily solved—problem of obtaining optimum mixing of air and oil with a minimum of noise throughout the whole spectrum.

Eliminating pulsations does not pose quite such a dilemma. The authors' suggestion of using a higher fan pressure is a good one. We have made a number of different experiments all of which show that as the pressure drop in the blast tube is increased the tendency to pulsate is diminished.

We have also had good luck with opening up vent holes around the outside of the blast tube to vent off the pressure surges before they get big enough to cause trouble—lowering the Q of the system as the authors say. At the moment this is the most practical way known for remedying a pulsing burner in the field.

The results of the work at the discusser's laboratory will be published within the next few months. In fact, the part dealing with the effect of fuel composition on noise has already been completed and will be submitted soon to the Society. By the way, it was concluded that the composition of the heating oil had no effect on the tendency of the flame to either pulsate or rumble.

EUGENE O. OLSON, West Des Moines, Ia.: My comments concern the mixing of air and oil, and I would like to refer to the authors' Fig. 5.

I would like to just briefly refer to some work done at Penn State under the title "Induced Air Flow in Fuel Sprays" by Dr. Hikmet Binark, who is now at the University of Istanbul in Turkey, and Dr. W. E. Ranz, who is now at the University of Minnesota. They did this work in connection with their studies of fuel sprays, and the results have a connection with this phenomenon of internal and external ignition.

In their work they found that with a hollow cone type of spray in still air the spray would draw air in from outside the spray envelope. This air enters the spray envelope at right angles to the envelope, beginning at a point where the droplet velocity is low enough to permit such air entering. After this air is inside the spray envelope, it changes direction and proceeds parallel to the spray axis. The extent of this infiltration or *inspiration* of air is such that it provides a fuel-air ratio very near to stoichiometric in the center of the spray. In this work they do not show what happens with a fan supplying air or with recirculation in a combustion chamber.

When changing to a solid cone nozzle, the inspiration of the air by the spray is the same. Air enters at right angles to the outer envelope of spray and changes direction inside the spray, the same as with the hollow cone nozzle. Because of the spray present in the center of the spray, however, the ratio of fuel to air is naturally far above stoichiometric.

I mention this because it ties in with reports from Battelle on the performance of hollow cone and solid cone sprays in connection with their studies on burner pulsation and oscillation. They found that rich mixtures in the center of the spray with solid cone nozzles provided a region of unstable ignition in the center of the spray.

This also ties in with experience in the field since it has been found by many people here present that this rumbling in oil-fired installations may be reduced by changing from solid cone nozzles to hollow cone nozzles. The noise is also reduced somewhat by using wider angle sprays which simply has the effect of giving less spray concentration in the center of the spray.

AUTHORS' CLOSURE (Mr. Sanders): With regard to Mr. Myler's comments concerning larger and shorter flue passages and a smaller barometric damper, the thermal efficiency must be considered. It was found that proper oil burner design could result in the use of from 5 to 10 percent excess air instead of the presently used 50 percent range. This reduction in excess is the factor which allows the change in flue and stack configuration. It should also be pointed out that perhaps more attention should be paid to insuring that the minimum damper opening is used in commercial units. It is felt that sometimes this factor may not receive the attention it needs for a minimum noise system.

The wording mentioned in Mr. Mittendorff's first question should have been fuel-air mixture as he indicates. The dotted curve of Fig. 3 indicates the relative pressure characteristics when the system is in pulsation as compared with the solid curve which shows the conditions during normal combustion.

In a hot fire box most of the flame is maintained by the hot refractory. There is, however, a portion of the combustion which takes place off the end of the electrodes. This is the portion which may oscillate and help cause rough burning. When the furnace is first started, the entire flame is sustained by the spark for a period of time. Here also there is a chance for spark-excited oscillation. The terms d-c and a-c airflow are used in the same manner as in electrical terminology where d-c refers to uni-directional flow and a-c to oscillatory flow. The symbol SPL in Figs. 7 and 8 is used to represent Sound Pressure Level.

Mr. Popyk's interest in what has been done to use the information found in this investigation was expected and omitted from the paper because it may be of specific proprietary value to the sponsoring organization.

The comments by Messrs. Speich, Putnam and Locklin are very appropriate and helpful especially because of the almost parallel program completed at Battelle.

Armour's investigation did not show vortex shedding to be responsible for exciting the furnace into resonance. It is felt that since vortex shedding frequency is a function of burner configuration and air velocity, there should be shedding even when the burner

is fired outside the furnace. Measurements made of the noise when the burner was fired in a room without a furnace did not reveal any peak noise frequencies as would have been expected from a vortex phenomenon.

The choice of 100 psi in the discussion on the reduction of pulsation was unfortunate and misleading. The numbers were selected as a means of showing the effect of a small fluctuation on a large pressure drop orifice, not as an indication that 100 psi would be required or desirable in the experimental program. It was found that pressure drops of as little as 2 to 3 in. of water were sufficient to make pulsation almost impossible in a conventional furnace.

The same reference level does apply directly to Figs. 7 and 8. However, the furnaces used for each curve were different and no correlation between those 2 curves is inferred. The data shown in Figs. 11 and 12 were obtained during combustion and represent levels at the damper opening. Equation 2 can be found in *Acoustics* by Beranek, (p. 135) as well as several other standard acoustic texts. Equation 3 gives the acoustic resistance due to flow in acoustic ohms and may be used with any acoustic resonant equation to determine the attenuation of a resonant system.

Dr. Ranz's findings as disclosed by Mr. Olson fit in very well with the findings of this present program. The entry of air into the fuel pattern away from the nozzle leaves an area nearly void of air in the center of the fuel spray near the nozzle. This area has very unstable mixing and combustion. Therefore, it probably is material in helping to generate the low frequency noise.

It is gratifying to see such similar results come from Mr. Sage. It will be of great interest to see Mr. Sage's paper on this subject in the near future.

1659

No. 1659

OMNI-DIRECTIONAL LOW-SPEED HOT-WIRE ANEMOMETER

By J. F. KEMP*, PRETORIA, SOUTH AFRICA

IN VENTILATION studies it is often required to measure flow speeds only while the actual flow direction is of little or no importance. Study of air speed distribution in a ventilated room may be quoted as a typical example.

Instruments commonly used in ventilation practice, such as the vane and the hot-wire types of anemometer, are generally not very suitable for measurements of this nature for these instruments are sensitive to flow direction, and their calibration depends on the attitude in which their velocity sensing elements are set up. Thus, even if the flow direction can be determined by means of smoke or tufts, an adjustment of the sensing element to comply with flow direction introduces errors as a result of the sensitiveness of the element to attitude.

Although the hot-wire anemometer described in this paper is not perfectly omni-directional, there is a fairly wide range of flow directions which does not affect its response.

BASIC CONSIDERATIONS

The hot-wire was selected as the basic type of sensing element for the development of an omni-directional instrument because of its high sensitivity at low speeds and of the variety of possibilities presented in its construction.

If heat losses from the element by natural convection are assumed negligible in comparison with the losses due to forced convection, the logical shape of the element for an omni-directional instrument is spherical. An additional requirement is that the heating of this spherical element should be of a symmetric nature, so that, at any given velocity, the same flow pattern and the same temperature distribution will exist over the exposed surface of the element, irrespective of the flow direction.

Unfortunately, heat exchange by natural convection becomes important as the flow speeds decrease. Since the natural convection currents always tend vertically upwards, they may be either assisted or opposed by the main flow, depending on the direction of flow. Natural convection therefore has an adverse effect on the

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* Senior Research Officer, National Mechanical Engineering Research Institute, South African Council for Scientific and Industrial Research.

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otherwise symmetric flow and heat transfer conditions, and consequently on the omni-directional properties of the instrument.

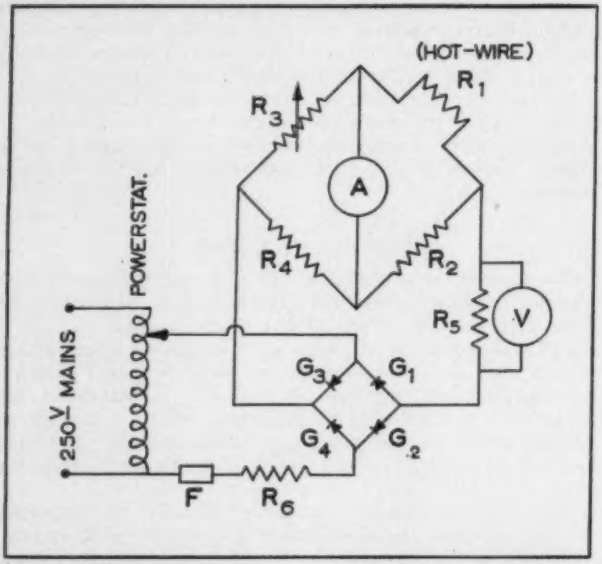
In order to limit the strength of the convection currents it is desirable to operate the heated sphere at a fairly low temperature. Too low a temperature, on the other hand, is disadvantageous to the sensitivity of the anemometer, and also aggravates the effect that ambient temperature variations have on the calibration of the instrument.

DESIGN AND CONSTRUCTION

Operation: The constant-resistance method of measurement was adopted for the anemometer. In this method the resistance of the hot-wire element is maintained constant by adjusting the heating current, which serves as a measure of the air speed.

The Callendar system, described by Ower[1], is used. The hot-wire element R_1 (30 S.W.G. iron wire) forms part of a Wheatstone bridge (Fig. 1), and has a resistance of 4 ohms when unheated. The other components of the bridge, R_2,

[1] Exponent numerals refer to References.



A = Micro-ammeter 100-0-100 microamps.
V = Voltmeter, d-c, 0-20 volts.
F = Fuse, ¼ amp.
R_1 = Hot-wire, 30 S.W.G. enamel-coated iron (approximately 4 ohms unheated).
R_2 = 400 ohms, wire-wound, constantan.
R_3 = 30 S.W.G. constantan wire.
R_4 = 100 ohms, wire-wound, constantan.
R_5 = 2 ohms, platinum wire, 0.001 in. diam.
R_6 = 200 ohms.
$G_1 - G_4$ = Metal rectifiers, type Q2BISG.M5, 1 amp, 200 volt.

FIG. 1....CIRCUIT DIAGRAM OF ANEMOMETER

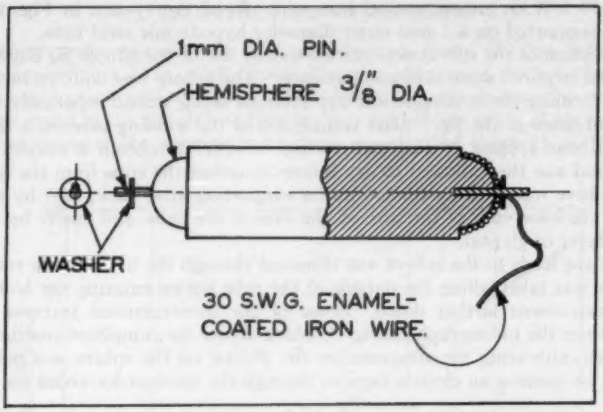


FIG. 2. . . .JIG FOR WINDING SPHERICAL HOT-WIRE ELEMENT

R_3 and R_4, are made of constantan wire. R_2 and R_4 have resistances of 400 and 100 ohms respectively. The effective length of R_3 is adjusted to yield balance of the bridge when a current of about 0.13 amps is passed through R_1, when R_1 is kept in still air.

The combination of R_5 and V is termed a volt-thermometer. R_5 comprises a 5-in. length of 0.001-in. diameter platinum wire, which is suitably shielded from air disturbances. As a result of the relatively high temperature coefficient of resistivity of platinum (0.003 per C deg between 0 and 100 C), the voltage drop across R_5 increases at a much higher rate than the heating current as the wire temperature rises.

From the relative values of the bridge resistors R_1, R_2, R_3 and R_4, it is evident that the greater portion of the current through R_5 passes through the hot-wire element R_1.

Whereas the air speed varies, over a wide range, approximately as the fourth power of the heating current through R_1, it varies by some power between 2 and 3 of the voltage drop across R_5, by virtue of the heating effect of the platinum wire. Thus, by utilizing the volt-thermometer instead of measuring directly the heating current, an appreciable gain in sensitivity and accuracy is achieved at higher flow speeds.

A simple rectifier bridge, comprising four 1-amp, 200-volt metal rectifiers, G_1, G_2, G_3, and G_4, is incorporated to permit a-c operation of the anemometer. The full-wave rectified current supplied to the Wheatstone bridge, though not filtered, enables either a d-c ammeter or a sensitive galvanometer to be used for balancing.

A-c power is supplied from the mains via a variable autotransformer (powerstat), which is used to regulate the heating current. By incorporating resistor R_6 (nichrome wire, 200 ohms) the powerstat is made to operate in the higher range of its voltage scale, so that current control can be effected more accurately. Fuse F protects the system against overloads.

Velocity Sensing Element: The velocity sensing element of the anemometer is made in the form of a $\frac{3}{8}$-in. diameter hollow sphere, whose walls consist of a single

layer of 30 S.W.G. enamel-coated iron wire (R_1 of the system in Fig. 1). The sphere is supported on a 1 mm outer diameter hypodermic steel tube.

Construction of the sphere was carried out by use of the simple jig illustrated in Fig. 2, and required some skill and patience. The sphere was built up in 2 halves, the wire forming the hemispherical cup sections being wound separately on the 2 half-round noses of the jig. After completion of the winding process, a thin layer of glyptal was applied to each cup section in order to obtain a smooth surface. The glyptal was then allowed to dry before removing the cups from the jig.

The sphere was then assembled on its supporting steel tube, first by soldering together the loose ends of the wire on the rims of the cups, and finally by applying another layer of glyptal.

One of the leads to the sphere was threaded through the hypodermic tube, while the other was taken along the outside of the tube before entering the hollow stem of the instrument farther down. Some of the constructional features may be studied from the photograph in Fig. 3, which shows the completed instrument.

Prior to calibrating the anemometer, the glyptal on the sphere was baked hard and dry, by passing an electric current through the element for about an hour.

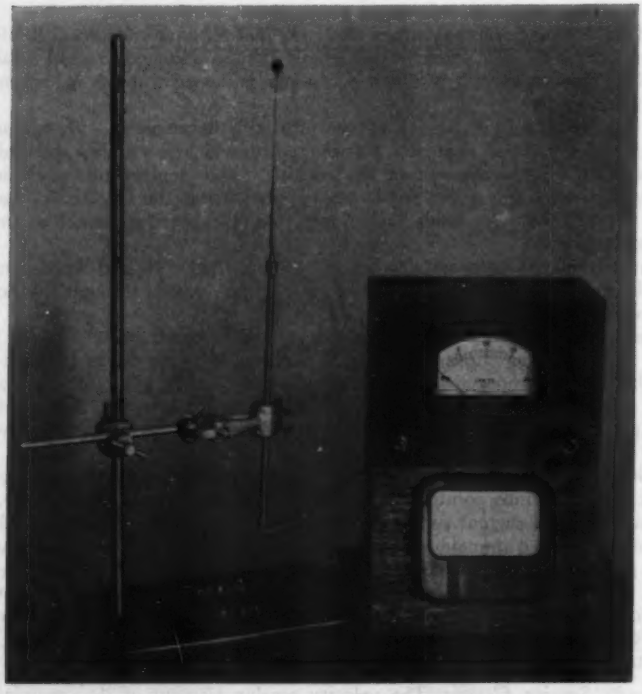


FIG. 3. . . .PHOTOGRAPH OF ANEMOMETER

CALIBRATION AND PERFORMANCE

The anemometer was carefully calibrated between 0 and 600 fpm against a low speed indicator[2] in a 24-in. wind-tunnel, the stem support of the hot-wire element being set up in a vertical attitude. The calibration curve is depicted in Fig. 4.

The air speed in the tunnel was then set to a convenient value of about 450 fpm and the response of the anemometer to yaw investigated every 15 deg from 0 to

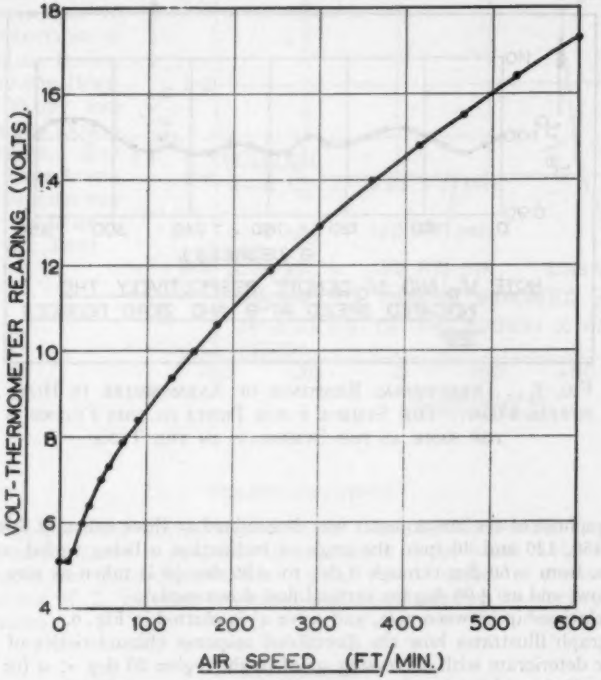


FIG. 4....CALIBRATION CURVE OF INSTRUMENT

360. The results of this test are presented in Fig. 5 where the ratio v_t/v_o is plotted against t, the angle of yaw (v_t and v_o denote respectively the indicated speeds at t and at zero degrees).

The maximum deviation of v_t/v_o from the value 1.00 at $t = 0$, is ± 2 percent. It may, therefore, be concluded that the instrument is 98 percent insensitive to the direction of flow in a horizontal plane.

Finally, the directional response of the anemometer was investigated for variations of the air velocity in a vertical plane. For this purpose a simple test rig was constructed. A small centrifugal fan was used to force air through a perspex

pipe, 4 ft long and 6 in. in diameter, gauze screens being provided to improve the flow quality in the pipe. The spherical element of the anemometer was set up centrally in the free jet near the pipe outlet. The rig was supported in such a way that it could be inclined to any desired position in a vertical plane, while the relative position between the spherical element of the anemometer and the pipe outlet remained unchanged.

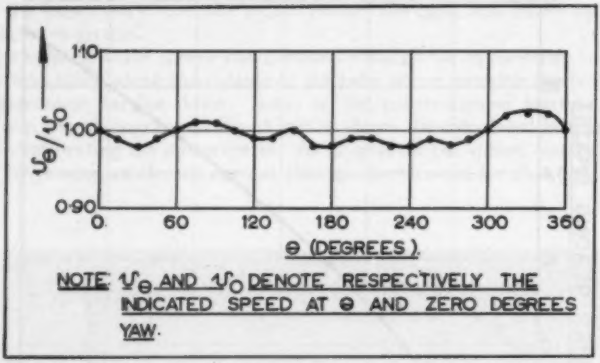


FIG. 5. . . . DIRECTIONAL RESPONSE OF ANEMOMETER IN HORIZONTAL FLOW. THE SYMBOL v SUB THETA IN THIS FIGURE IS THE SAME AS THE SYMBOL v_t IN THE TEXT

The response of the anemometer was determined at three constant flow speeds, *viz.*, at 450, 120 and 30 fpm, the angle of inclination α being varied, at 10 deg intervals, from -60 deg through 0 deg to $+90$ deg (α is taken as zero for horizontal flow, and as $+90$ deg for vertical flow downwards).

The relationship between v_α/v_0 and α (or a) is plotted in Fig. 6.

The graph illustrates how the directional response characteristics of the anemometer deteriorate with decreasing speed in the region 50 deg $< \alpha$ (or a) < 90 deg. At 30 fpm the natural convection currents appear to have approximately the same strength as the main flow. When the flow is directed downwards, the convection currents are forced back towards the sphere, thereby restricting the rate of heat escape.

At higher speeds this effect is less marked, as forced convection gradually becomes the predominating influence whereby heat exchange takes place between sphere and air current.

The set of curves in Fig. 6 indicates that the anemometer is approximately 90 percent insensitive to flow direction in the range -50 deg $< \alpha$ (or a) $< +50$ deg.

Apart from its omni-directional properties, other noteworthy features of the instrument are its simplicity of construction and operation, the small dimensions of its velocity sensing element, and a fairly quick reaction time of less than one minute.

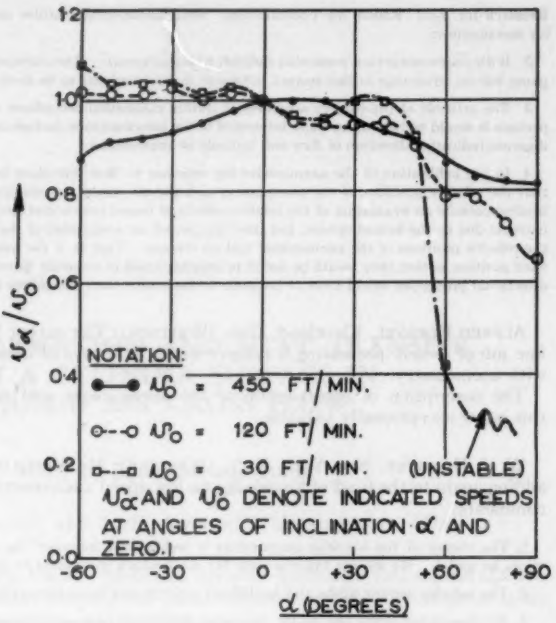


FIG. 6. . . .RESPONSE OF ANEMOMETER TO FLOW DIRECTIONS IN A VERTICAL PLANE. α IS ZERO FOR HORIZONTAL FLOW, AND POSITIVE FOR DOWNWARD FLOW. THE SYMBOL v SUB ALPHA IN THIS FIGURE IS THE SAME AS THE SYMBOL v_α IN THE TEXT

ACKNOWLEDGMENTS

The work described was performed in the National Mechanical Engineering Research Institute and is published by permission of the South African Council for Scientific and Industrial Research. The author wishes to thank B. van Wamelen and M. L. Strydom for valuable assistance during the development of the instrument.

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[1] E. Ower: *The Measurement of Air Flow* (Chapman and Hall, London, 1949, p. 249, 3rd ed.).

[2] J. F. Kemp: A sensitive aerodynamic drag balance as an indicator of low air speeds (*Journal of Scientific Instruments*, October 1957, p. 411).

DISCUSSION

D. R. BAHNFLETH, Chicago, Ill., (WRITTEN): In connection with this excellent paper, there are several points to be mentioned since they could properly be called to the attention of the author:

1. The anemometer described in the subject paper is not too different in principle from an anemometer developed at the University of Illinois and described in a Society paper entitled Room Air Distribution

Research for Year 'Round Air Conditioning. Both anemometers utilize small heated sensing elements for measurement.

2. Both anemometers are somewhat difficult to manufacture. The anemometer described in the subject paper has an advantage in this respect, although it would appear to be more susceptible to damage.

3 The attitude of the subject anemometer during calibration procedures is not too clearly stated, and perhaps it would be helpful for those interested in the information to include on Figs 4, 5 and 6 a schematic diagram indicating direction of flow and attitude of anemometer.

4. In the calibration of the anemometer for response to flow directions in a vertical plane, it appears that the relative positions of the anemometer and the air stream remain unchanged. This type of calibration permits an evaluation of the relative effects of forced convection currents and natural convection currents due to the heated sphere, but does not permit an evaluation of the response due to a change in the relative positions of the anemometer and air stream. That is, if the anemometers were installed in a fixed position so that they would be useful in investigations of room-air distribution, further study of their directional properties would have to be made to determine their suitability before such use.

ALFRED KOESTEL, Cleveland, Ohio (WRITTEN): The author has again done his usual fine job of clearly presenting a subject which is always of interest to engineers dealing with anemometry.

The description of construction of the anemometer and calibration results makes this paper exceptionally valuable.

W. O. HUEBNER, New York, N. Y. (WRITTEN): Mr. Kemp is to be complimented for adding again to the fund of knowledge on low-speed anemometry. I have the following comments:

1. The theory of the hot-wire anemometer is well-known, however, the equation for the instrument might be useful. We are not familiar with Mr. Callendar's instrument in this country

2. The velocity sensing probe and its difficult construction merit further detailed discussion.

3. An illustration of the test rig for measuring directional response is desirable.

AUTHOR'S CLOSURE: A point that evidently needs to be clarified is the set-up and method used for determining the directional response of the anemometer to flow directions in a vertical plane.

The small test rig mentioned in the paper was supported in such a manner that the 6-in. perspex pipe, together with the fan, could be inclined at various angles α to the horizontal. The horizontal axis, about which the unit pivoted, lay on the extension of the pipe center line, some distance downstream of the outlet, and at right angles to the center line. The anemometer was set up in a fixed position, with its supporting stem in its normal, vertical attitude, and with the spherical sensing element located on the intersection between the pipe center line and the axis about which the pipe pivoted. Thus the sensing element always remained at a constant distance from the outlet and in the same region of the free jet, irrespective of the inclination of the jet.

The method permitted a realistic evaluation of the directional response to inclined flows.

Mr. Bahnfleth suggests that the instrument appears to be rather frail. This is true, but most hot-wire anemometers are subject to this criticism. One possibility of overcoming this problem may be to shield the sensing element by means of a light wire cage. This may lead to a clumsy design, however, for a cage of reasonable size may well impair the directional response of the instrument.

In reply to the first point raised by Mr. Huebner, it must be stated that no attempt was made to develop a basic equation for the instrument. For this purpose it would be necessary to establish the heat transfer properties of a spherical shell in an airstream. If a simple relationship can be derived between the Nusselt number and the flow Reynolds number, it may be possible to assess the effect of variations in ambient temperature on the accuracy of the anemometer.

1660

No. 1660

FROST FORMATION AND HEAT TRANSFER ON A CYLINDER SURFACE IN HUMID AIR CROSS FLOW

Part I—Experimental Study

By P. M. CHUNG* AND A. B. ALGREN**, MINNEAPOLIS, MINN.

THE HEAT and mass transfer processes between humid air and a solid surface kept at a subfreezing temperature have become common phenomena with application of low temperature refrigeration.

Major existing investigations of the problem of heat transfer and frost formation pertinent to the present study are included in References 1 through 14.

The investigations reported in many of the references were concerned with the overall effect of an evaporator frosting on the performance of a refrigeration system. Experimental performance data for a frosted evaporator coil are given in some of the papers, whereas information obtained through experiences in application are found in others.

In several of the References (1, 6, 9, 10, 11, 13 and 14) the basic analytical studies on heat transfer and frost formation were conducted both theoretically and experimentally. The particular problems analyzed in the referenced papers included free convection, and forced convection over a flat plate.

In 1933, Piening[9] carried out one of the first analytical studies, by forming ice on a cylinder from a natural convection current of humid air. Since then, several people, listed in the references, studied the mechanism of heat transfer accompanied by frost formation essentially on a flat plate.

The latest study on the subject is found in the work of Sugawara and his associates.[14] Frost was formed on a 15-cm × 100-cm brine-chilled flat plate as a stream of controlled humid air passed over it. Air velocities of 1.5- to 7.1-meters per sec were used. Typical data obtained from the experiment are reproduced

* Assistant Professor, Department of Mechanical Engineering, University of Minnesota.
** Professor, Department of Mechanical Engineering, University of Minnesota. Member of ASHAE.
• Exponent numerals refer to References.
Presented at the 65th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, Pa., January 1959.

here in Fig. 1. It is seen from Fig. 1 that the total heat transfer rate decreased with time, and reached quasi-steady state in 60 to 80 min.

Virtually no basic analytical work in connection with forced convection across a tube has been reported.

For an engineering purpose, the transfer process between the humid air and a circular tube in cross flow is probably more important than the process over a flat plate. In the absence of previous work, this particular phase of the subject was studied.

EXPERIMENTAL SET-UP

Test Cylinder No. 1: It was decided in the heat transfer studies to evaporate a refrigerant in the test section rather than to pass chilled brine through it so that

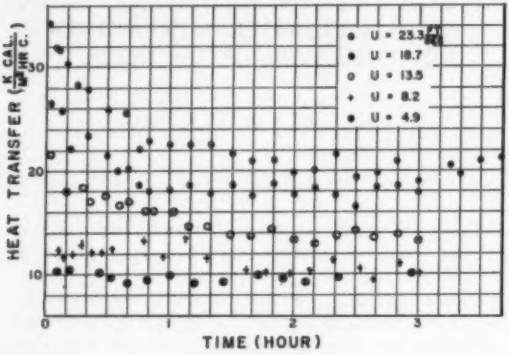


FIG. 1. . . .VARIATION OF TOTAL HEAT TRANSFER WITH TIME, REPRODUCED FROM REFERENCE 14

uniform temperatures may be obtained on the test surface without sacrificing accuracy of measurement. Measurement of mass of refrigerant evaporated per unit time then can be used to calculate heat transfer.

Test cylinder No. 1 was constructed with the previously stated principle in mind in order to measure local heat transfer along the circumference of the cylinder for any given time. Actual design and construction were quite complicated. Fig. 2 is a sketch of the internal structure of the cylinder. The cylinder which is 1⅛ in. OD and 2 ft in overall length, is made of copper, and divided into test and guard sections. The test surface is 1 ft long and has surface area of 0.0273 sq ft. The test section and guard section are separated from each other by 0.1-in. air gap all the way around except at the cylinder surface. A $^1/_{16}$-in. diam rubber O-ring was split in half and fitted into the gap along the borderline between the test and guard surfaces. Wax was then poured over it to make it tight. The rubber ring and wax were fitted into the gap to a depth of about $^1/_{16}$ in. and the excess portion above the cylinder surface was filed to a smooth surface. The test and guard sections have separate refrigerant passages in and out.

Four thermocouples were led into the test cylinder, 2 from each end. One set was soldered to the test section and the other to the guard section. To measure the temperature of the test surface accurately, the thermocouples should be connected to the surface. However, it was impossible practically to pass the thermo-

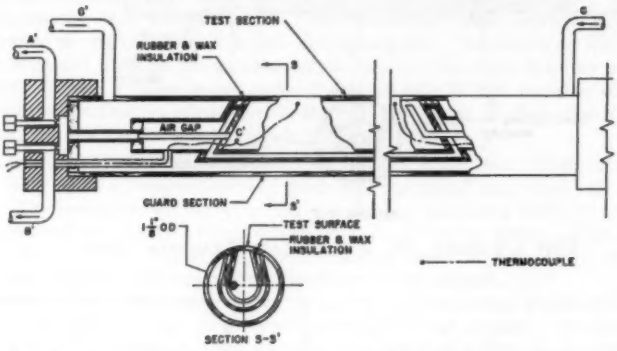


FIG. 2.... TEST CYLINDER NO. 1 FOR HEAT TRANSFER TEST

couples through the bottom of the test section and reach the test surface from below. The thermocouples, therefore, were soldered to the side of the test section exposed to the air gap. It was thought, that since the thermocouples were located very close to the test surface and the surface would be in direct contact with the

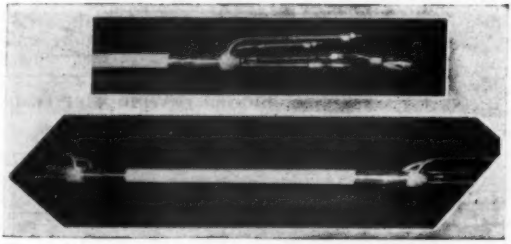


FIG. 3.... PHOTOS SHOWING THE ASSEMBLED CYLINDER NO. 1

evaporating refrigerant liquid during the tests, the temperature of the test section measured would represent the test surface temperature without appreciable error. A very thin coat of white enamel was put on the surface of the cylinder to lessen the error in heat transfer measurement which might be caused by variation of surface emissivity with frost formation. The white enamel was known to have an emissivity of 0.92 whereas the emissivity of frost is 0.98 (Reference 15). Possible

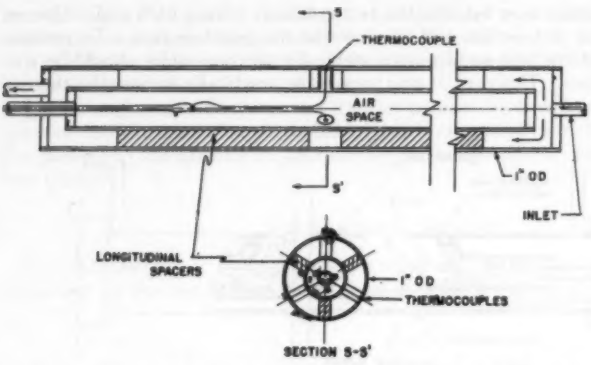


FIG. 4. . . . TEST CYLINDER NO. 2 FOR MASS TRANSFER TEST

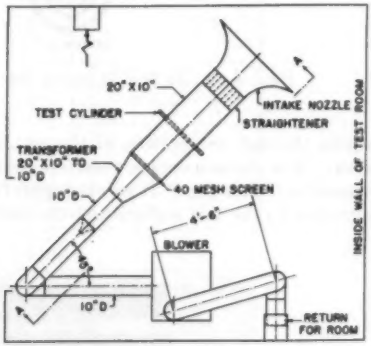


FIG. 5. . . . WIND TUNNEL IN TEST ROOM

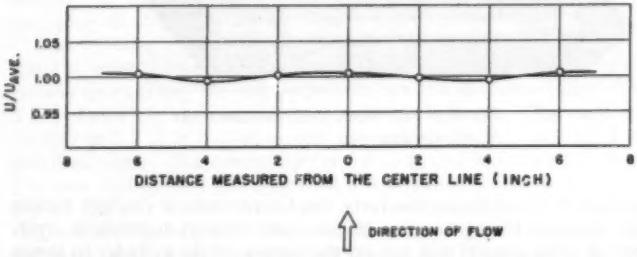


FIG. 6. . . . TYPICAL VELOCITY DISTRIBUTION IN THE TEST SECTION OF THE WIND TUNNEL (FOR AVERAGE AIR VELOCITY OF 15.27 FT PER SEC)

effect of the enamel on frost formation was tested and found to be negligible. A photograph of the assembled cylinder No. 1 is shown as Fig. 3.

Test Cylinder No. 2: The second cylinder was made to measure the mass transferred to the surface. Cylinder No. 2 measured 1 in. OD and 2 ft in overall length. Fig. 4 is a sketch of the cylinder structure.

Three thermocouples were led into the cylinder from one end, and each one was connected to the test surface. The 3 thermocouples were located along the circumference of the cylinder at equal distances apart. As shown in Fig. 4, refrigerant is passed through the annular space between the outer cylinder and the inner tube. Three dividers were placed longitudinally in the annular space to insure uniform distribution of refrigerant through the space. A thin coat of white enamel was put on the completed cylinder similar to No. 1.

Test Room and Wind Tunnel: A low-speed wind tunnel was constructed and installed (Fig. 5) in a temperature and humidity controlled test room (15- × 16- × 9-ft). The inside of the tunnel at the test section measured 10 × 20 in. Air flow was created by a turbo-blower driven by a 5-hp electric motor. The blower was rated for 1300 cfm delivery against a static pressure of 1 in. Hg. Air flow rate was controlled by a damper at the discharge of the blower.

Air sucked through the intake nozzle from the test room was discharged from the blower directly into one of 2 return plenums in the rooms. The air, conditioned for both temperature and humidity was returned to the room uniformly through the perforated ceiling air supply panel by a separate blower. Smoke was generated in the test room and its motion, observed during a test, showed that the air turbulence level in the room was not higher than that in an average room.

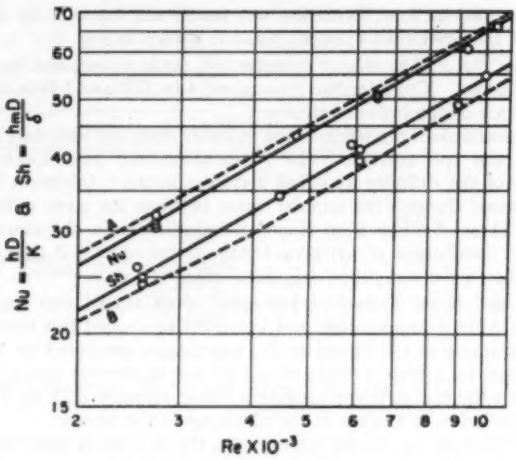
MEASUREMENTS

Free Stream Velocities at the Test Section: At higher air velocities, pitot tube traverses were made across the test section. Pressures were measured with a micromanometer with vernier scale attachment graduated to 0.001 in. of water. Fig. 6 shows the typical velocity profile across the test section. The average free stream velocities were obtained as a function of the corresponding pressure drop across a 4-in. flow nozzle located downstream.

For lower air velocities the local velocities at the test section could not be measured accurately. However, the 4-in. flow nozzle used in the experiment had been calibrated previously and very accurate calibration data of air flow *vs* pressure drop across the nozzle were available. Therefore, measured values of the mass flow of air were used in the following manner to calculate the free stream velocity at the test section. The simplified equation (Equation 1) is given in Reference 16 for the boundary layer development along the surface of a body with an arbitrary pressure gradient.

$$\frac{U_s l_i^*}{\frac{\mu}{\rho}} = \frac{0.47}{\left(\frac{U_s}{U_o}\right)^5} \int_o^x \left(\frac{U_s}{U_o}\right)^5 dx \qquad (1)$$

Equation 1 was integrated numerically from the beginning of the intake nozzle to the test section for a given mass flow rate. This yielded the momentum thickness l_i at the test section. Solution of a boundary layer equation at $du_s/dx = 0$ such as at the test section gave $l^*/l_i = 2.554$ (Reference 16). Therefore, the displacement thickness l^* was obtained from l_i. Each surface at the test section



[A] Nu NUMBER OBTAINED BY COMINGS ET AL (19) FOR TURBULENCE INTENSITY LESS THAN 3%

[B] Nu NUMBER OBTAINED BY HILPERT (33)

FIG. 7....NUSSELT (NU) AND SHERWOOD (SH) NUMBERS OBTAINED FOR BARE CYLINDER (τ = 0)

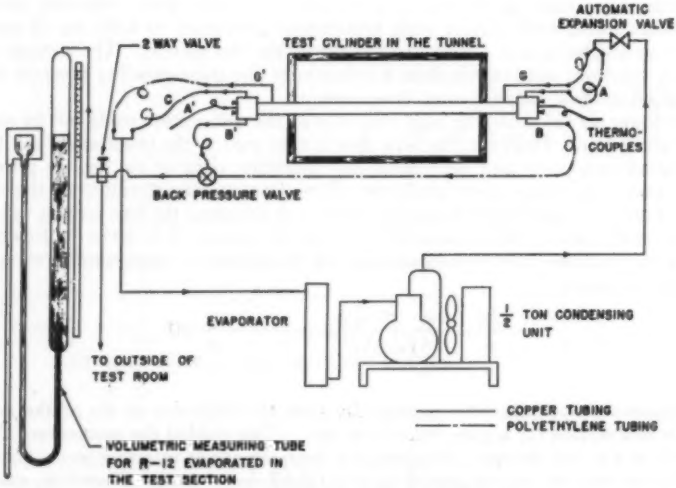


FIG. 8....HEAT TRANSFER TEST SET-UP

FIG. 9. . . .PHOTO SHOWING MEASURING GLASS-TUBE IN POSITION IN TEST ROOM

was then assumed to have moved inward by l^*. Finally, by definition of displacement thickness, a free stream velocity for a given measured mass flow was obtained by dividing the total volume flow by the corresponding decreased hypothetical cross-sectional area at the test section.

Stream velocities thus obtained were further corrected to cover the blockage created by the test cylinder. Equation 2 (found in Reference 17) was used for the blockage correction.

$$U_0 = U_0'(1+\varepsilon) \qquad (2)$$

and

$$\varepsilon = \frac{\pi^2}{12}\left(\frac{A_f}{A_T}\right)$$

It was shown earlier[18, 19] that the turbulence in the free air stream could influence the transfer process considerably. The turbulence intensity at the test section was found to be 1 to 3 percent in the experiment of Reference 19 when air was brought into the tunnel without any artificial upstream disturbances. Since the wind tunnel employed for the present study was similar in physical construction to that used previously,[19] it was thought that the turbulence level in the present wind tunnel would be about 1 to 3 percent also. This assumption was shown to be correct as the Nusselt numbers obtained from the present experiment checked closely with those found previously[19] for turbulence intensities less than 3 percent. (See Fig. 7).

Heat Transfer to Test Cylinder No. 1: A one-half horsepower condensing unit installed in the test room was used to refrigerate the test cylinders. R 12 was used as the refrigerant throughout the entire experiment. The R 12 flow circuit shown in Fig. 8 was used in connection with the volume measuring glass-tube shown as Fig. 9, to obtain the heat transfer measurement. As the R 12 vapor entered the glass tube from the top, the overflow type cup connected to the glass tube by a plastic tube was lowered manually. The length of the water column

displaced in the glass tube by the R 12 vapor was read to 1 mm by the attached scale.

In a continuous operation the automatic expansion valve (Fig. 8) kept the evaporating temperature of the refrigerant at a constant value both in the test section and the guard section. The refrigerant entered the guard section at G and left at G' while it entered the test section at A and left at C.

As the time for the heat transfer measurement neared, the refrigerant supply to the test section was stopped by closing the needle valve A. The needle valve C

NOMENCLATURE

x = distance measured in the direction of air flow.
U_e = velocity outside the boundary layer in the direction of air flow.
U_∞ = free stream velocity after blockage correction.
U'_∞ = free stream velocity before blockage correction.
t = temperature, in general, or at the frost-air interface.
w = specific humidity, in general or at the frost-air interface.
t_∞ = free stream temperature.
W_∞ = free stream specific humidity.
l_1 = momentum thickness of the boundary layer.
l^* = displacement thickness of the boundary layer.
A_f = frontal area of the test cylinder.
A_T = free cross-sectional area of the wind tunnel at the test section.
V_2 = volume of water displaced per unit time in the measuring glass tube.
v_2 = specific volume of R-12 vapor in the measuring glass tube.
v_{1f} = specific volume of saturated liquid in the test section.
v_1 = specific volume of saturated vapor in the test section.
L = latent heat of sublimation of the frost.
L_v = latent heat of R 12 at the test section.
Re = Reynolds number.
Pr = Prandtl number.
Sc = Schmidt number.
Nu = Nusselt number.
Sh = Sherwood number = $h_m D_o/\delta$.
St_m = Stanton number for mass transfer = Sh/S_cR_e.
St = Stanton number for heat transfer = N_u/P_rR_e.
D_o = diameter of the bare cylinder.
μ/ρ = kinematic viscosity.
ρ = density.
δ = mass diffusivity.
C_p = specific heat at constant pressure.
τ = time.
θ = defined in Fig. 10.
h = heat transfer coefficient.
h_m = mass transfer coefficient.

Subscripts

v = vapor.
a = air.
w = at the solid surface.

was also closed and the one at B' was opened. The back-pressure valve was previously set to keep the pressure upstream at the same value as the operating pressure in the test cylinder. The evaporating refrigerant vapor from the test section of the cylinder came through the back-pressure valve and was led out of the test room through a two-way valve opened to the outside. During this operation, which usually took less than a minute, the evaporating temperature in the test section was undisturbed since at no time was the pressure in the section varied. Measurement of the evaporation rate in the test section began with the opening of the two-way valve to the measuring glass tube and, at the same time, starting of a timer. Sufficient volume of the refrigerant vapor usually collected in 30 to 90 sec. At the end of the measurement the two-way valve was opened to the outside and the timer was stopped. The previous steps with the needle valves were reversed to return the test section to the original circuit.

All connections to the test cylinder were polyethylene flexible tubes. This allowed the free rotation of the cylinder necessary to obtain local heat transfer along the circumference of the cylinder. Because of the symmetry between the upper and lower halves of the cylinder, the heat transfer was measured only along the surface of the lower half.

Heat transfer to the test section was evaluated from the volumetric measurement of the evaporating refrigerant by Equation 3; thus

$$\text{Heat Transfer} = \frac{\frac{V_2}{V_1}}{1-\frac{V_{1f}}{V_1}} L_v \qquad (3)$$

The test cylinder was also calibrated for heat leakage between the test and guard sections.

Mass Transfer to Test Cylinder No. 2: A rectangular tray (10.064- × 1.75-in.) was constructed for measuring frost collected on the cylinder. The tray was made of 0.01 in. thick berylium copper. Both ends of the tray were cut out in a semi-circle so that the ends would fit the cylinder exactly. A lid was made in a similar manner to fit over the tray. After a predetermined period for a run, the test was stopped and the tray and the lid were pushed tightly against the cylinder until the sharp semi-circular ends cut into the frost and made contact with the cylinder surface. The portion of the cylinder surface which represented the test section was the area 10.064 in. in length which was covered by the tray and lid. After the frost was melted by raising the cylinder temperature the tray was removed, and the water film still on the cylinder surface was absorbed thoroughly and quickly by chemical filter papers and placed in the tray. The tray containing the water and filter papers was then weighed on a commercial balance and measured to 0.001 grams accurately. The net weight of the frost was obtained, finally from the gross weight.

Accuracy of the Measurements: All temperatures were measured with soldered thermocouples of 24-gage copper and constantan wires previously calibrated to 0.1 F deg accuracy for the temperature range encountered. A precision portable potentiometer was used for temperature readings. With the instruments used, all temperatures obtained were estimated to be within the accuracy of 0.1 F deg.

Maximum fluctuations allowed through each test, which usually lasted 3 hours, were as follows: temperatures at test cylinder No. 1, ±0.2 F deg; temperatures at test cylinder No. 2, ±0.1 F deg; free air stream temperatures, ±1 F deg; room humidity, ±2 percent.

Test Series: The heat transfer and mass transfer experiments were conducted for the following conditions.

Series 1: Air stream temperature = 60 F; air stream specific humidity = 0.00427 lbv/lba; test cylinder surface temperature = 0 F; Reynolds numbers for heat transfer = 2720, 4710, 6500, 9270, and 10630; Reynolds numbers for mass transfer = 2592, 4430, 6020, 8920, and 9965; sensible heat ratio = 0.775.

Series 2: Air stream temperature = 70 F; air stream specific humidity = 0.0040 lbv/lba; test cylinder surface temperature = −10 F; Reynolds numbers for heat transfer = 2,720 and 6,500; Reynolds numbers for mass transfer = 2,592 and 6,070; sensible heat ratio = 0.817.

TABLE 1....TYPICAL HEAT TRANSFER DATA

Re = 9270; θ = 30°; Barometric pressure = 14.25 psia; Average W_o = 0.0043 lb_v/lb_a

TIME MIN	MEASURING GLASS TUBE READING CM PER MIN	TEMPERATURE, FAHRENHEIT				HEAT LEAKAGE CORRECTION, BTU/MIN	SPEC. VOL. R 12 GAS COLLECTED CU FT PER LB	LATENT HEAT OF VAPORIZATION IN TEST SECTION, BTU/LB	TOTAL HEAT MEASURED, BTU PER MIN	NET HEAT TRANSFER, BTU PER HR
		IN TEST SECTION	IN GUARD SECTION	IN FREE AIR STREAM	VAPOR IN GLASS CYL					
5.0	70.4	0.45	0.85	59.0	60.1	−0.0272	3.17	68.71	0.494	29.5
15.0	61.4	−0.10	−0.17	59.0	60.2	0.0048	3.17	68.77	0.431	26.2
30.5	60.6	0.08	−0.25	59.9	60.8	0.0224	3.17	68.77	0.426	26.9
45.3	54.8	−0.23	−0.40	60.0	61.2	0.0088	3.18	68.79	0.413	25.3
69.1	54.9	−0.13	−0.28	59.9	61.0	0.0010	3.18	68.77	0.385	23.2
95.5	53.3	−0.20	−0.50	60.3	61.0	0.0204	3.18	68.78	0.374	23.7
109.0	51.2	−0.24	−0.40	60.1	61.0	0.0109	3.18	68.78	0.359	22.2
146.0	53.3	−0.25	−0.31	60.6	61.5	0.0041	3.18	68.78	0.374	22.7

Series 3: Air stream temperature = 70 F; air stream specific humidity = 0.0084 lbv/lba; test cylinder surface temperature = 10 F; Reynolds numbers—same as series 2; sensible heat ratio = 0.625.

FORMULATION OF THE DATA

Heat Transfer: A typical set of test data and corresponding calculated heat transfer values are shown in Table 1. Heat transfer values are also plotted in Fig. 10. Values shown in both the table and graph do not include correction for radiation heat transfer. To obtain the heat transfer for the bare cylinder, the

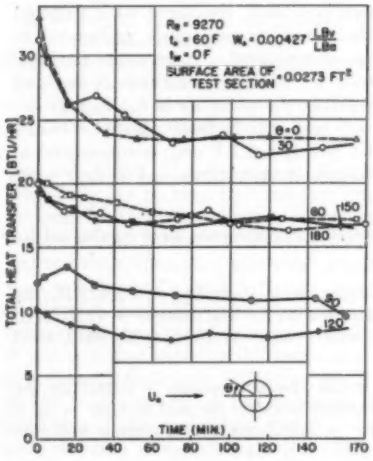


FIG. 10....TOTAL HEAT TRANSFER MEASURED ALONG CIRCUMFERENCE OF CYLINDER NO. 1

first 3 test points were used and the values for zero time were extrapolated. The average heat transfer for the entire cylinder is of greater interest here than the local values along the circumference. The heat transfer for the entire cylinder was obtained by summing up the 7 local values for any given time with proper regard to test areas. The average heat transfer values for the cylinder were then corrected for radiation. In the radiation correction, it is necessary to know the air-frost interface temperature. The total heat transfer for a given air velocity was momentarily assumed to be a function only of the temperature difference between the air stream and the interface at any time. The hypothetical interface temperatures were then deduced from the total heat transfer measured at different times and the radiation corrections made. The possible error in the heat transfer values obtained due to the lack of accuracy in the radiation correction was estimated to be about 1 to 3 percent of the total heat transfer.

If the sensible heat associated with the vapor is neglected, the total heat transfer, Q_T, is:

$$Q_T = (Sh)\rho\pi\delta\left[\left(\frac{Nu}{Sh}\right)\left(\frac{Sc}{Pr}\right)(t_0 - t)Cp + L(w_0 - w)\right]$$

For the range of present study:

$$(Nu/Sh)(Sc/Pr) = 1.136 \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(See Fig. 7)}$$

A total heat transfer coefficient h_e in Btu per (hour) (square foot) (Btu per pound) was defined so that:

$$Q_t/\pi D_o = h_e(e_o - e_w)$$

where

$$e = 1.136\ C_p t + LW$$

TABLE 2. . . .TYPICAL TOTAL HEAT TRANSFER NUMBER

Re = 9270; W_o = 0.00427 lb_v/lb_a; t_o = 60 F; t_w = 0 F

TIME, MIN	HEAT TRANSFER, BTU PER HOUR			$\frac{h_e D_o}{\rho\delta}$
	TOTAL TO CYLINDER Q'_T	BY RADIATION, Q_r	TOTAL CORRECTED, Q_T	
0	219.0	14.3	204.7	51.3
10	206.0	13.4	192.6	48.2
20	197.5	13.0	184.5	46.2
40	189.2	12.4	176.8	44.2
60	181.5	11.8	169.7	42.5
80	179.0	11.7	167.3	41.8
100	178.2	11.6	166.6	41.7
120	175.2	11.4	163.8	41.0
140	176.0	11.5	164.5	41.1
160	174.0	11.4	162.6	40.7

The total heat transfer number is then defined as $h_e D_o/\rho\delta$ and for a bare cylinder this is equal to Sherwood number.

A typical tabulation of the number is shown in Table 2. Graphical representations are given in Figs. 11 and 12 for the entire series of tests. In addition, the total heat transfer at zero time was divided into sensible and latent heat transfer and the Nusselt number was calculated from the sensible heat transfer and shown in Fig. 7. The Sherwood number in the figure was obtained directly from the mass transfer test. All the property values used in the analysis were obtained from References 20 and 21.

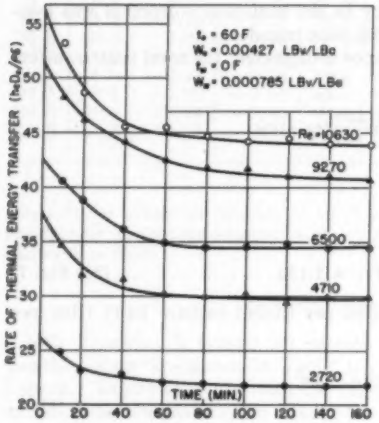


FIG. 11. . . . VARIATION OF TOTAL HEAT TRANSFER RATE WITH TIME

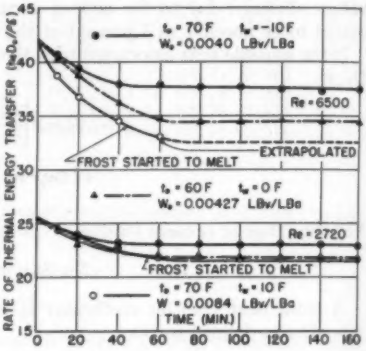

t_a = 70 F t_w = -10 F
W_a = 0.0040 LBw/LBa
Re = 6500
EXTRAPOLATED
FROST STARTED TO MELT
t_a = 60 F t_w = 0 F
W_a = 0.00427 LBw/LBa
Re = 2720
FROST STARTED TO MELT
t_a = 70 F t_w = 10 F
W = 0.0084 LBw/LBa
TIME (MIN.)
RATE OF THERMAL ENERGY TRANSFER ($h_e D_o/\rho\delta$)

FIG. 12. . . . VARIATION OF TOTAL HEAT TRANSFER WITH TIME WITH REYNOLDS (RE) NUMBER = 2720 AND 6500

Mass Transfer: Figs. 13 and 14 show the mass transfer as a function of time. The slopes for each curve at different times represent the instantaneous mass transfer rates.

DISCUSSION OF THE RESULTS

In Fig. 7, the Nusselt numbers obtained for the bare cylinder of the present study agree closely with those given[19] for turbulence intensity of less than 3 percent. This was predicted at the beginning from the fact that the wind tunnel set-up used was similar to that employed in the reference.

The ratio of the Stanton numbers St_m/St is equal to $(Pr/Sc)^{2/3}$ according to Colburn's analogy between heat and mass transfer. Then for the range of the present experiments, $St_m/St = (Pr/Sc)^{2/3} = 1.036$. The actual ratio of the Stanton numbers, however, is obtained from Fig. 7 to be 0.88. It was found in earlier work[11] that the Stanton number ratio was 0.90 when a heat transfer and sublimation experiment was conducted on a frosted flat plate with the properties

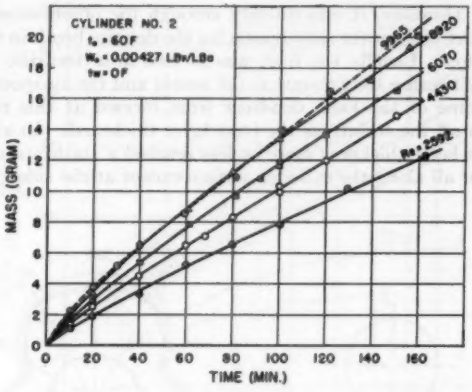


FIG. 13. . . .TOTAL MASS TRANSFER CYLINDER NO. 2, (TEST CONDITIONS AS LISTED)

of air similar to those used here. Therefore the ratio obtained here agrees with that obtained in the reference.

As a test was started a uniform layer of frost usually soon formed along the entire surface of the cylinder. Fastest growth of the frost was usually observed near the stagnation point. In about 100 min the thickness of the layer was more or less evened out for the whole cylinder except near the sides of the cylinder (see Fig. 15). A definite discontinuity in the thickness and density of the frost was observed near this point. The mass transfer coefficient is minimal at this region

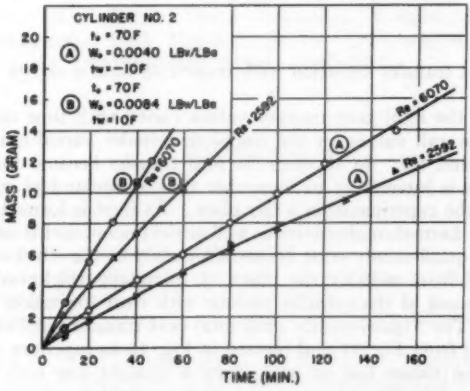


FIG. 14. . . .TOTAL MASS TRANSFER WITH CYLINDER NO. 2 (TEST CONDITIONS AS LISTED)

for a cylinder. However, it was thought through the observations that the low transfer coefficient was not the only reason for the definite break in the frost formation at this region. Usually the frost was formed in a dendritic manner in this region, probably because the pressure is the lowest and the air speed high. It was thought that some of the loose dendritic frost formed at this region was continuously blown off the surface as the frost layer thickened. In all the tests conducted the frost layer thickness more or less reached a static value of 0.1 to 0.15 in. in about 2 hr all along the cylinder surface except at the sides.

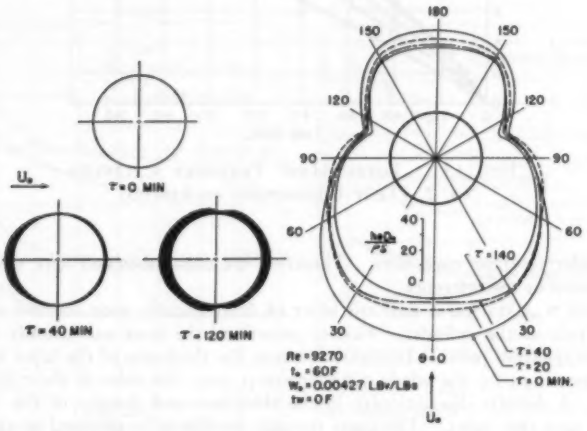
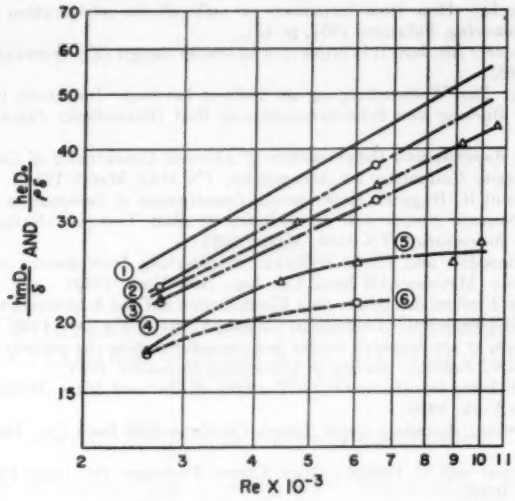


FIG. 15. . . .FORMATION OF FROST AND TOTAL HEAT TRANSFER ALONG THE CIRCUMFERENCE OF THE CYLINDER

A typical heat transfer variation with respect to time is shown in Figs. 10 and 15.

Even though the local heat transfer values varied with time rather irregularly (Fig. 10), the overall values for the complete cylinder varied quite orderly with time, (Figs. 11 and 12). In all cases the heat transfer became quasi-steady in 60 to 100 min. It is interesting to remember that a similar fact was observed by Sugawara[14] in the experiment on a flat plate. As frost is formed on the surface, the density and thermal conductivity as well as the thickness of frost is continuously changed. The quasi-steady state is reached mainly as the effect of the increasing conductivity of frost nullifies the effect of increasing thickness of frost. The increased roughness of the cylinder surface with frost formation also affects the heat transfer. The 5 quasi-steady state total heat transfer numbers for test *Series* 1 were obtained from Fig. 11 and plotted in Fig. 16 as functions of the Reynolds numbers. These values line up well along a straight line with logarithmic coordinates. Line 3 in Fig. 16 represents these particular values. *Series* 2 and 3 were tested at 2 Reynolds numbers only. A straight line was drawn through the 2 test points for *Series* 2 and 3 in the light of *Series* 1.



① $Sh = \frac{h_m D}{\delta} = \frac{heD}{\rho\delta}$ FOR $T = 0$

② $\left(\frac{heD_v}{\rho\delta}\right)$ FOR $t_a = 70F$ $W_a = 0.0040$ $\frac{LBv}{LBa}$ $tw = -10F$

③ $\left(\frac{heD_v}{\rho\delta}\right)$ FOR $t_a = 60F$ $W_a = 0.00427$ $\frac{LBv}{LBa}$ $tw = 0F$

④ $\left(\frac{heD_v}{\rho\delta}\right)$ FOR $t_a = 70F$ $W_a = 0.0084$ $\frac{LBv}{LBa}$ $tw = 10F$

⑤ $Sh' = \frac{h_m D_v}{\delta}$ FOR $t_a = 60F$ $W_a = 0.00427$ $\frac{LBv}{LBa}$ $tw = 0F$

⑥ $Sh' = \frac{h_m D_v}{\delta}$ FOR $t_a = 70F$ $W_a = 0.0040$ $\frac{LBv}{LBa}$ $tw = -10F$

FIG. 16. . . .TOTAL HEAT TRANSFER NUMBER AND SHERWOOD (SH) NUMBER AFTER EQUILIBRIUM IS ESTABLISHED (TIME > 80 MIN.)

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1661

No. 1661

FROST FORMATION AND HEAT TRANSFER ON A CYLINDER SURFACE IN HUMID AIR CROSS FLOW

Part II: Theoretical Study and Conclusions

By P. M. CHUNG* AND A. B. ALGREN**, MINNEAPOLIS, MINN.

IN A FLOW of air normal to the axis of a circular cylinder, a laminar boundary layer exists along the surface of the approach side up to the boundary layer separation point. The boundary layer becomes turbulent just before the separation at high Reynolds Numbers. For most engineering work only the laminar boundary layer exists, and it separates approximately at 70 deg measured from the stagnation point. Following the separation, the Karman Vortex wake is formed[15].

Mathematical analyses of flow and heat transfer, as well as the experimental analysis, have been carried out by several persons along the laminar boundary layer. However, only the experimental data are available for the surface of the cylinder downstream from the separation point. The following 3 boundary layer integral equations are available for steady-state flow, heat transfer, and mass transfer respectively, along the laminar boundary layer.

$$U_o\frac{dU_o}{dx}(l^*+2l_i)+U_o^2\frac{dl_i}{dx}-V_wU_o=\frac{\mu}{\rho}\left(\frac{dU}{dy}\right)_w \quad\cdots\cdots(1)$$

$$\frac{d}{dx}\int_o^{t_o}U(t_o-t)dy=\alpha\left(\frac{dt}{dy}\right)_w+V_w(t_o-t_w) \quad\cdots\cdots(2)$$

$$\frac{d}{dx}\int_o^{t_o}U(W_o-W)dy=\delta\left(\frac{dW}{dy}\right)_w+V_w(W_o-W_w) \quad\cdots(3)$$

In Equation 1, μ/ρ is the kinematic viscosity, and l^* and l_i are the displacement and the momentum thickness respectively.

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** Professor, Department of Mechanical Engineering, University of Minnesota. Member of ASHAE
[15] Exponent numerals 22 to 33 refer to References at end of this paper; exponent numerals 21 and lesser refer to References in Part I (see p. 227).
Presented at the 65th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, January 1959.

Equations 1 and 3 were solved[22] for the evaporation of water, and for the general process of diffusion from a flat plate to a gas stream[23]. They were also solved for a stagnation point of a sphere in the same paper. All three Equations 1, 2 and 3 were solved in connection with the frosting flat plate[14]. In all these solutions the Karman-Pohlhausen-Kroujiline method[24] was used. Accurate solution of the 3 equations for the laminar region of a circular cylinder is possible, but is tremendously complicated and time consuming. Moreover, the steady-state solution thus ob-

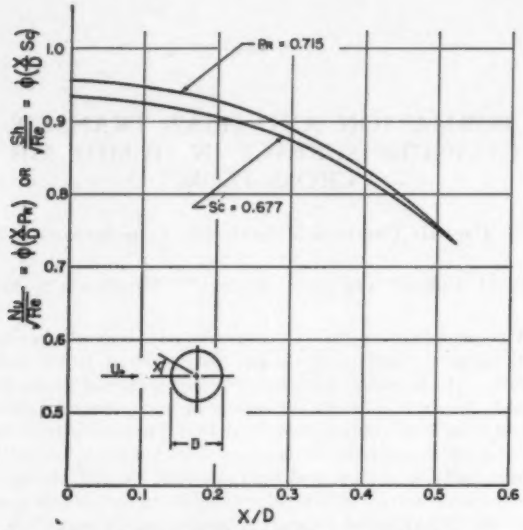


FIG. 17....THEORETICAL VALUES OF NUSSELT AND SHERWOOD NUMBERS ALONG THE LAMINAR REGION OF THE CYLINDER (CALCULATED BY THE METHOD IN REFERENCE 24)

tained will not be of much value for the problem of frost formation which is essentially a transient situation.

Without solving the 3 equations directly, the following studies were made to evaluate the heat transfer along the laminar boundary layer with mass transfer.

The concentration of water vapor involved in the process of frost formation is small. Therefore, the magnitude of V_w is also small and can be neglected in the solution of the equations without appreciable error. Several approximate methods of solving Equations 1 and 2 for $V_w = 0$ have been devised by various people for different cylinder profiles. In References 24 and 25 the now well-known method of the equivalent wedge-type solution of a heat transfer problem is presented. This method utilizes the existing solutions along wedges with the flow represented by the equation $U_\infty = (\text{constant}) X^m$. The heat transfer along the laminar region of a circular cylinder was solved for $A = 0.70$.[24] The Prandtl number is found to be 0.715

for the condition of the air in the present study. The sensible heat transfer coefficients along the cylinder surface are calculated here for Pr = 0.715 with the help of the pre-drawn curves found in Reference 24. These values are shown in Fig. 17.†

For $V_w = 0$, Equation 2 for the heat transfer and Equation 3 for the mass transfer are similar. The solutions of the equations, hence, also could be expected to be similar. Through the dimensional analysis with Equations 2 and 3, and through numerous experiments (References 21 and 26 through 31), the following general relations are known today:

$$\text{Nu} = f_1 (\text{Pr}, \text{Re}), \text{ for heat transfer, and}$$

$$\text{Sh} = f_2 (\text{Sc}, \text{Re}), \text{ for mass transfer.}$$

The functions f_1 and f_2 are more or less the same. For geometrically similar cases f_1 is equal to f_2 if Pr = Sc. Generally, however, it is also extended to cases where Pr is not exactly equal to Sc.

For the small concentrations of water vapor and temperature difference involved in the present study, either the partial pressure or concentration difference of water vapor may be used as the driving force in mass transfer.

The general method of approach to the problem of simultaneous transfer of heat and mass was discussed[21] and this general method, with some modifications, is applied here to the laminar region of the circular cylinder. The principle of heat and mass transfer analogy, and previously calculated values of heat transfer coefficients shown in Fig. 17 aided in derivation of the following equations.

There exist velocity, temperature, and partial pressure gradients in the direction normal to the surface near the surface. For the sensible heat transfer at the surface:

$$K\left(\frac{\partial T}{\partial y}\right)_w = h(T_o - T_w) \qquad (4)$$

Now,

$$\frac{Nu}{\sqrt{Re}} = \phi\left(\frac{x}{D}, Pr\right) = \phi\left(\frac{x}{D}, 0.715\right)$$ was calculated in Fig. 17.

Therefore Equation 4 becomes:

$$K\left(\frac{\partial T}{\partial y}\right)_w = \frac{K}{D}\sqrt{Re}\,\phi\left(\frac{x}{D}, Pr\right)(T_o - T_w) \qquad (5)$$

From the heat and mass transfer analogy:

$$Sh = f_2(Sc, Re) = f_1(Sc, Re), \text{ therefore}$$

$$\frac{Sh}{\sqrt{Re}} = \phi\left(\frac{x}{D}, Sc\right).$$

At the average temperature of the air at 30 F encountered here, Sc = 0.677. Therefore $\text{Sh}/\sqrt{\text{Re}} = \Phi\ (x/D\ 0.677)$ was also calculated by the equivalent wedge method, and is presented in Fig. 17.

For the mass transfer at the surface with the aid of Fick's Law:

$$\left(\frac{\partial C}{\partial y}\right)_w = \left(\frac{\sqrt{Re}}{D}\right)\left(\frac{1}{R_v T}\right)\phi\left(\frac{x}{D}, Sc\right)(P_o - P_w) \qquad (6)$$

† Figure numbers in Part II of this paper are numbered 17 to 30, inclusive; Figs. 1 to 16, inclusive, are in Part I (see p. 213).

It is known that as the diffusion of water vapor proceeds through the laminar boundary layer, a secondary convective current is set up. This phenomenon is analyzed[21] and the study of this secondary current yields:

$$V_w = m_w R_v T_w / P_t \quad \ldots\ldots\ldots\ldots \quad (7)$$

$$\text{and} \quad \left(\frac{\partial C}{\partial y}\right)_w = \frac{m_w}{\delta}\left(1-\frac{P_w}{P_t}\right) \quad \ldots\ldots\ldots\ldots (8)$$

and Equation 5 becomes:

$$m_w\left(\frac{x}{D}\right) = C_m\left(\frac{x}{D}\right)(P_0 - P_w) \quad \ldots\ldots\ldots\ldots (9)$$

$$\text{where} \quad C_m\left(\frac{x}{D}\right) = \left(\frac{\delta}{DR_vT}\right)\sqrt{Re}\left[\frac{\phi(x/D, Sc)}{1-(P_w/P_t)}\right] \quad (10)$$

The total heat transfer per unit area along the laminar boundary layer is:

$$q_t\left(\frac{x}{D}\right) = K\left(\frac{\partial t}{\partial y}\right)_w + V_w(t_0 - t_w)C_p\rho + m_w L \quad \ldots\ldots (11)$$

$$\text{now} \quad \frac{R_v T_w \rho}{P_t} = \frac{M}{M_v} \quad \ldots\ldots\ldots\ldots (12)$$

Substitution of Equations 5, 7, 9 and 12 into Equation 11 yields Equation 13, thus

$$q_t\left(\frac{x}{D}\right) = C_u\left(\frac{x}{D}\right)(T_0 - T_w) \quad \ldots\ldots\ldots\ldots (13)$$

$$\text{where,} \quad C_u\left(\frac{x}{D}\right) = \frac{K}{D}\sqrt{Re}\,\phi\left(\frac{x}{D}, Pr\right) + C_m\left(\frac{x}{D}\right)\left[\frac{M}{M_v}C_p + \frac{L}{T_0 - T_w}\right](P_0 - P_w) \quad \ldots\ldots\ldots\ldots (14)$$

The preceding analysis does not apply for the surface of the cylinder downstream from the boundary layer separation point.

Through extended studies, Colburn[26–30] established the general analogy between heat and mass transfer processes as stated in Equation 15, thus

$$St_m/St = (Pr/Sc)^{2/3} \quad \ldots\ldots\ldots\ldots \quad (15)$$

The Stanton number ratio was calculated and was compared to the experimental value for the entire cylinder in Part I.

VARIATION OF TOTAL HEAT TRANSFER WITH TIME

Here, the transient process of heat transfer effected by the growth of the frost layer on the cylinder surface is studied. The process here is extremely complicated. Since the heat and the mass transfer coefficients are not uniform over the cylinder surface, each position along the surface will go through different mode of variation of conditions with time. Strictly, the cylinder ceases to be a circular one as frost is being formed on its surface (See Fig. 15, Part I). The Reynolds number and thus the transfer coefficients also vary with the change in the geometry of the cylinder profile. In addition, the density and the thermal conductivity of the

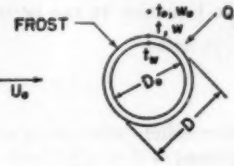


FIG. 18. . . .DIAGRAM OF ORIGINAL CYLINDER AND QUANTITIES USED IN SETTING UP HEAT BALANCE

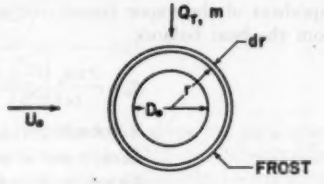


FIG. 19. . . .DIAGRAM USED IN SETTING UP MASS BALANCE

PART II NOMENCLATURE

The following symbols are used in this paper (Part II) in addition to the Nomenclature in Part I (see p. 220):

y = distance measured normal to the cylinder surface.
U_o = free stream velocity.
T = absolute temperature, in general, or at the frost-air interface.
W = specific humidity, in general, or at the frost-air interface.
D = diameter of the frosted cylinder.
r = radius of the frosted cylinder.
τ = time.
m = mass transfer.
q, Q = heat transfer.
C = concentration of vapor.
K = thermal conductivity.
α = thermal diffusivity.
μ/ρ = kinematic viscosity; μ = absolute viscosity.
h = coefficient of heat transfer.
h_m = coefficient of mass transfer.
V = velocity in y direction.
R = gas constant.
he = total heat transfer coefficient defined by; Q_t/Surface area = he $(e_o - e_w)$.
e = 1.136 $C_p t + LW$ . . . (See Part I).
M = molecular weight of mixture.
P = partial pressure of vapor.
l_t = thickness of thermal boundary layer.
l_m = thickness of diffusion boundary layer.
t = temperature, Fahrenheit degrees.
$\phi(x/D, P_r)$ = a particular function of x/D, and P_r.
$n, C_1, C_2, C_3, a, b,$ = constants defined in the text.
(x/D) in mw (x/D), C_m (x/D), etc. . . . shows that mw, C_m etc. are functions of x/D.

Subscripts

o = free stream conditions.
w = surface conditions of the bare cylinder.
f = frost.
V = vapor.
T, = total value.
x means in x direction.

frost vary unpredictably and through a wide range. It is clear that no mathematical solution to the problem is possible if all the numerous variables were considered.

In the present study the problem is idealized as follows: In Fig. 18, D_0 is the diameter of the original cylinder. It is assumed that frost forms uniformly along the cylinder surface and that the conductivity of frost does not vary with time. Two major time variables are considered in the transient analysis, *viz*: (a) variation of air-frost interface temperature with time; and (b) variation of Reynolds number and heat and mass transfer coefficients with the growing frost layer.

Let t_i represent the interface temperature and D the outer diameter of the frosted cylinder for any time, τ. Then if the heat storage within the frost layer and the

superheat of the vapor transferred are neglected, the Equation 16 can be set up from the heat balance.

$$Q_T = \frac{2\pi K_f (t - t_w)}{\ln(D/D_0)}$$

$$= h\pi D(t_0 - t) + L\pi D h_m \rho (W_0 - W) \text{-------(16)}$$

Here, t and w are functions of time.
Equation 16 can be transformed into:

$$\frac{2K_f}{K} \frac{(t - t_w)}{\ln(D/D_0)} = Nu(t_0 - t) + Sh\left(\frac{Pr}{Sc}\right)\frac{L}{C_p}(W_0 - W)$$

----------(17)

From the heat and mass transfer analogy, Nu = C_1 (Sh).

$$From\ Fig.\ 7,\ Sh = C_2\left(\frac{U_0 D}{\mu/\rho}\right)^n = C_2\left(\frac{U_0 D_0}{\mu/\rho}\right)^n \left(\frac{D}{D_0}\right)^n$$

$$Nu = C_3 (Re)^n \left(\frac{D}{D_0}\right)^n,$$

$$where\ C_3 = C_1 C_2\ and\ Re = \left(\frac{U_0 D_0}{\mu/\rho}\right)$$

Write: $W_0 - W = (W_0 - W_w) - (W - W_w)$, and $T_0 - T = (T_0 - T_w) - (T - T_w)$. Here, W_w and W represent the specific humidities of the saturated air at t_w and t, respectively. Since $(t - t_w)$ is usually small it can be assumed that $T - T_w = b\,(W - W_w)$ where b is a constant within the particular temperature range.

Finally, Equation 17 transforms into Equation 18.

$$\frac{W - W_w}{W_0 - W_w} = \frac{C_3 \dfrac{t_0 - t_w}{W_0 - W_w} + C_2 \dfrac{Pr}{Sc}\dfrac{L}{C_p}}{2\dfrac{K_f}{K}\left(\dfrac{bRe^{-n}}{(D/D_0)^n \ln(D/D_0)}\right) + C_2 \dfrac{Pr}{Sc}\dfrac{L}{C_p} + C_3 b}$$

----------(18)

For the mass transfer process, Fig. 19 is considered, and the mass balance of Equation 19 is made on the cylinder.

$$2\pi r \rho_f dr = h_m 2\pi r \rho (W_0 - W) d\tau \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

This is changed to the following form.

$$\frac{d\left(\dfrac{D}{D_0}\right)}{d\left(\dfrac{U_0 \tau}{D_0}\right)} = \frac{2C_2(W_0 - W)}{Re^{1-n} Sc \left(\dfrac{\rho_f}{\rho}\right)\left(\dfrac{D}{D_0}\right)^{1-n}}$$

Writing $W_0 - W = (W_0 - W_w) - (W - W_w)$, the equation is transformed to the integral form shown in Equation 20.

$$\int_1^{D/D_0} f\left(\frac{D}{D_0}\right) d\left(\frac{D}{D_0}\right) = (W_0 - W_w)\frac{U_0 \tau}{D_0} \text{----------(20)}$$

$$\text{where, } f\left(\frac{D}{D_o}\right)=\frac{Re^{1-n}Sc\left(\frac{\rho_f}{\rho}\right)\left(\frac{D}{D_o}\right)^{1-n}}{2C_e\left[1-\frac{W-W_w}{W_o-W_w}\right]} \qquad \text{-------(21)}$$

For a given value of $(t_o - t_w)/(W_o - W_w)$, Equation 18 gives the parameter $(W - W_w)/(W_o - W_w)$ as a function of the diameter ratio (D/D_o). Substitution of $(W - W_w)/(W_o - W_w)$ into Equation 20 gives the relationship between the parameters D/D_o and $U_o\,\tau/D_o$.

Finally the relation between $(W - W_w)/(W_o - W_w)$ and $U_o\,\tau/D_o$ is obtained, and the total heat transfer at any given time is evaluated from Equation 16. The total mass transferred up to any time, τ is obtained from the relation between D/D_o and $U_o\,\tau/D_o$.

In the foregoing analysis, the integral in Equation 20 must be evaluated numerically since the integrand $\int (D/D_o)$ is complicated. In some applications, however, the temperature difference $(t_o - t_w)$ is small. The integration of $\int (D/D_o)$ becomes rather simple under such cases for the duration of time in which (D/D_o) is nearly equal to 1. For small values of $(t_o - t_w)$, $(t_o - t_w)/(W_o - W_w) = b = (t - t_w)/(W - W_w)$ and Equation 18 simplifies to Equation 22.

$$\frac{W-W_w}{W_o-W_w}=\frac{\psi_o}{\dfrac{2\dfrac{K_f}{K}bRe^{-n}}{(D/D_o)^n ln(D/D_o)}+\psi_o} \qquad \text{----------(22)}$$

$$\text{where, } \psi_o=C_3b+C_4(Pr/Sc)\frac{L}{C_p}$$

Now for the values of (D/D_o) close to one, $ln\,(D/D_o) \simeq (D/D_o) - 1$.

With this in mind, Equation 22 is substituted into Equation 21 and in turn Equation 21 is substituted into Equation 20. Equation 20 is then integrable, and its integration finally yields Equation 23.

$$(W_o-W_w)\frac{U_o\tau}{D_o}=\psi_1\left(\frac{\psi_o}{6}-\frac{\psi_2}{2-n}\right)+\psi_1\left[\frac{\psi_2}{2-n}\left(\frac{D}{D_o}\right)^{2-n}+\frac{\psi_o}{3}\left(\frac{D}{D_o}\right)^3-\frac{\psi_o}{2}\left(\frac{D}{D_o}\right)^2\right] \qquad \text{----------(23)}$$

where,

$$\psi_1=\frac{ScRe\left(\frac{\rho_f}{\rho}\right)}{4C_e b\left(\frac{K_f}{K}\right)}, \text{ and } \psi_2=2\left(\frac{K_f}{K}\right)bRe^{-n}$$

DIFFUSIONAL PROCESS WITHIN FROST LAYER AND VARIATION OF THE THERMAL CONDUCTIVITY WITH TIME

It is known that the properties of already-formed frost vary with time. The mode of variation depends on different parameters. It is thought that the variation of frost density and conductivity is largely due to the diffusion of vapor through

the porous frost layer. It is also thought that the migration of vapor occurs mainly due to its concentration gradient which exists through the frost. This concentration gradient is created by the temperature gradient which usually exists in the frost layer.

Fig. 20 represents the enlarged cross section of a porous frost layer.

The air in the frost is saturated with water vapor at the local temperature of the frost. Therefore, a particular concentration gradient exists for the given temperature gradient through the frost.

Consider the vapor which is to be driven across the plane at x. The vapor cannot go through the solid portion but must go through the gaps between the solids.

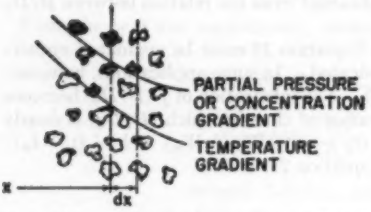


FIG. 20. . . . .REPRESENTATION OF ENLARGED CROSS-SECTION OF A POROUS FROST LAYER

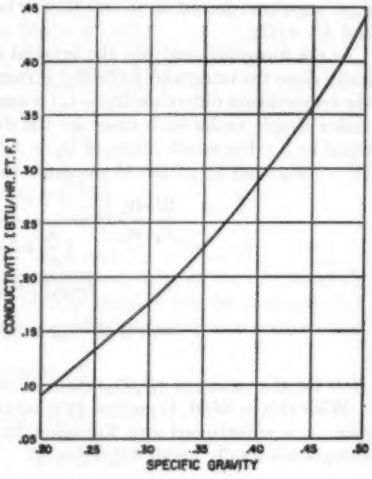


FIG. 21. . . . .VARIATION OF THERMAL CONDUCTIVITY OF FROST WITH SPECIFIC GRAVITY (OBTAINED BY DEVAUX, REFERENCE 10)

Therefore, the total free area available for the diffusion process is determined by the over-all density of the frost at the location. If there is no solid at all, the diffusion flux across x at any given time τ is represented by the equation $m_x = -\delta\,(\partial c/\partial x)$. On the other hand, let ρ_{fo} represent the density of the frost for which there is no free area available for the vapor migration. Then:

$$m_x = -\delta(1 - \rho_f/\rho_{fo})\,(\partial c/\partial x) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

where

$$\rho_f = \rho_f\,(x, \tau) \text{ and } C = C\,(x, \tau).$$

The total heat transfer at x for any given time τ is:

$$q_x = -K_f(\partial t/\partial x) - \delta L(1 - \rho_f/\rho_{fo})\,(\partial c/\partial x). \quad . \quad . \quad . \quad . \quad . \quad (25)$$

Now, for the saturated air and for the small temperature differences imposed on the frost, the temperature t can be represented as a linear function of C, therefore, $t = a + bc$.

It is seen in Fig. 21 that K_f can be approximately represented by a linear function of ρ_f within a limited range, therefore, $K_f = i + K_o \rho_f$ where i and K_o are constants. The numerical value of i is very small, therefore, $K_f = K_o \rho_f$. With these approximations imposed on Equation 25,

$$q_x = -\left[K_o b \rho_f + \delta L\left(1 - \frac{\rho_f}{\rho_{fo}}\right)\right]\frac{\partial C}{\partial x} \qquad (26)$$

$$q_{(x+dx)} = q_x - \left[K_o b \rho_f + \delta L\left(1 - \frac{\rho_f}{\rho_{fo}}\right)\right]\frac{\partial^2 C}{\partial x^2}\,dx - \left[K_o b - \frac{\delta L}{\rho_{fo}}\right]\frac{\partial C}{\partial x}\frac{\partial \rho_f}{\partial x}\,dx,$$

$$\text{and } q_x - q_{(x+dx)} = C_{pf}\rho_f b\frac{\partial C}{\partial \tau}$$

Finally,

$$\left(K_o b - \frac{\delta L}{\rho_o}\right)\rho_f\frac{\partial^2 C}{\partial x^2} + \delta L\frac{\partial^2 C}{\partial x^2} + \left(K_o b - \frac{\delta L}{\rho_o}\right)\left(\frac{\partial C}{\partial x}\right)\left(\frac{\partial \rho_f}{\partial x}\right) = C_{pf}\rho_f b\frac{\partial C}{\partial \tau} \qquad (27)$$

Similarly, from Equation 24, the transient diffusion Equation 28 is derived.

$$\frac{\partial^2 C}{\partial x^2} - \frac{\rho_f}{\rho_{fo}}\frac{\partial^2 C}{\partial x^2} - \frac{1}{\rho_{fo}}\frac{\partial C}{\partial x}\frac{\partial \rho_f}{\partial x} = \frac{1}{\delta}\frac{\partial \rho_f}{\partial \tau} \qquad (28)$$

Simultaneous solution of Equation 27 and 28 with proper boundary conditions yields $\rho_f(x, \tau)$ and whence $K_f = K_f(\rho_f) = K_f(x, \tau)$. Equations 27 and 28 are applicable to a frost layer formed on a flat plate.

A similar method was applied to a layer of frost formed on a circular cylinder and Equations 29 and 30 were derived:

$$\left(K_o b - \frac{\delta L}{\rho_{fo}}\right)\rho_f\frac{\partial^2 C}{\partial r^2} + \delta L\frac{\partial^2 C}{\partial r^2} + \left(K_o b - \frac{\delta L}{\rho_{fo}}\right)\frac{\partial C}{\partial r}\frac{\partial \rho_f}{\partial r} + \left(K_o b - \frac{\delta L}{\rho_{fo}}\right)\frac{\rho_f}{r}\frac{\partial C}{\partial r} + \delta L\frac{1}{r}\frac{\partial C}{\partial r} = C_{pf}\rho_f b\frac{\partial C}{\partial \tau} \qquad (29)$$

and,

$$\left(1 - \frac{\rho_f}{\rho_{fo}}\right)\frac{\partial^2 C}{\partial r^2} - \frac{1}{\rho_{fo}}\frac{\partial \rho_f}{\partial r}\frac{\partial C}{\partial r} + \left(1 - \frac{\rho_f}{\rho_{fo}}\right)\frac{1}{r}\frac{\partial C}{\partial r} = \frac{1}{\delta}\frac{\partial \rho_f}{\partial \tau} \qquad (30)$$

Both sets of the Equations 27 and 28, 29 and 30 are complicated, and could not be solved analytically. However, with some simplifications derived from the physical conditions, Equations 27 and 28 are solved here numerically.

The second term on the right hand side in Equation 26 represents the heat transfer due to the latent heat of the vapor migrated. In a thin frost layer, the vapor

migration is a slow process. The heat transfer, then associated with the vapor migration is of small magnitude compared to the heat transfer by conduction. Therefore this particular term in Equation 26 is neglected. Also, the heat stored

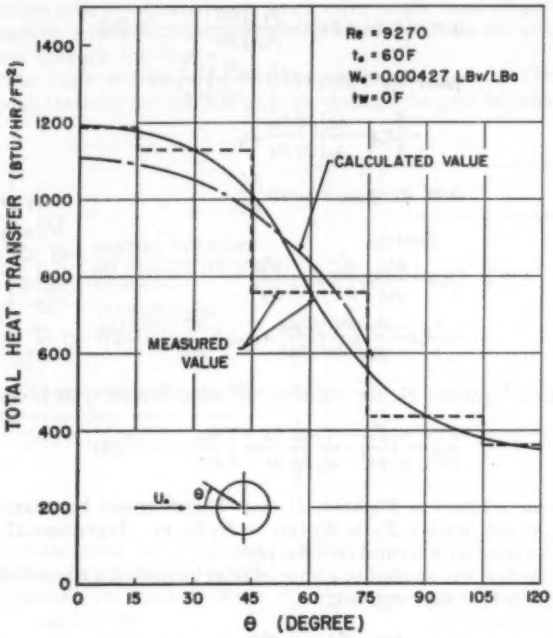


FIG. 22. . . .COMPARISON OF MEASURED AND CALCULATED VALUES OF TOTAL HEAT TRANSFER RATE ALONG THE LAMINAR BOUNDARY LAYER

in the frost can be neglected since usually the frost layer considered is thin. Under these assumptions, Equation 27 is simplified to:

$$K_o b \rho_f \frac{\partial^2 C}{\partial x^2} + K_o b \frac{\partial C}{\partial x} \cdot \frac{\partial \rho_f}{\partial x} = 0 \text{ ----------(31)}$$

Equations 28 and 31 can be solved numerically by changing them to a set of finite difference Equations.

NUMERICAL EVALUATION OF THE EQUATIONS

Equations 13 and 14 applied to the bare cylinder were solved for the boundary conditions at which the experiments were conducted in Part I. The total heat

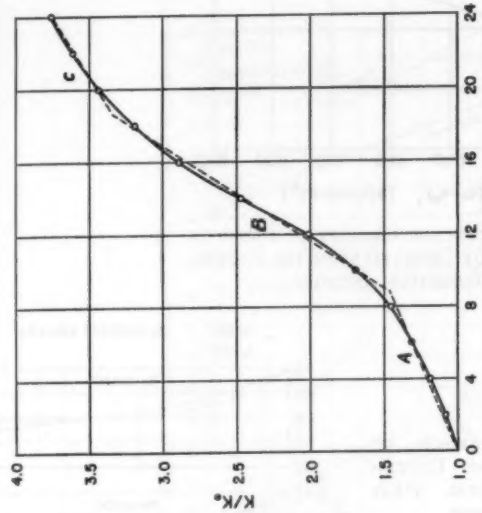


FIG. 24. . . . VARIATION OF OVERALL CONDUCTIVITY WITH TIME

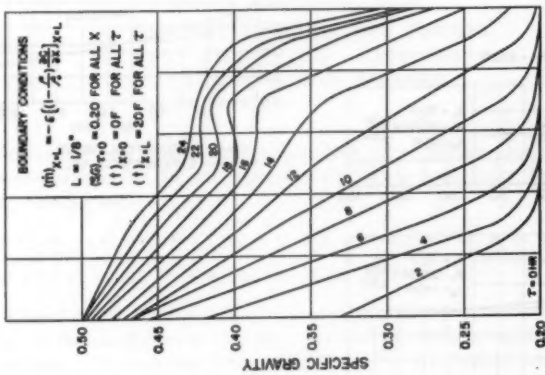

SPECIFIC GRAVITY
DISTANCE FROM WALL (INCH)
BOUNDARY CONDITIONS
$(\dot{m})_{x=L} = -\epsilon\{(1-\frac{\rho_2}{\rho_1})\frac{\partial G}{\partial x}\}_{x=L}$
L = 1/8"
$(G)_{\tau=0} = 0.20$ FOR ALL X
$(t)_{x=0} = 0F$ FOR ALL τ
$(t)_{x=L} = 20F$ FOR ALL τ

FIG. 23. . . . . VARIATION OF SPECIFIC GRAVITY WITH TIME

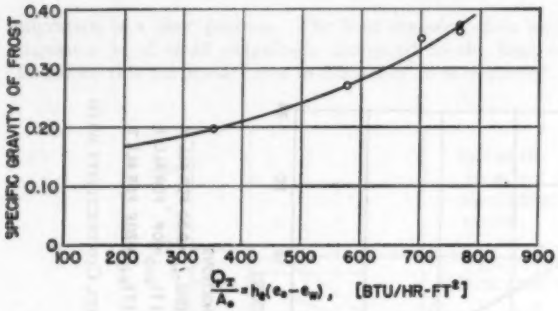
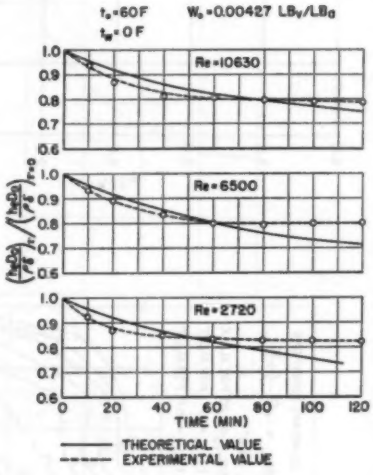


FIG. 25. . . .VARIATION OF SPECIFIC GRAVITY OF FROST VS TOTAL HEAT TRANSFER FLUX DEDUCED FROM THE EXPERIMENTAL AND THEORETICAL STUDIES

FIG. 26 (*right*). . . .COMPARISON OF THEORETICAL VALUES AND EXPERIMENTAL VALUES OF TOTAL HEAT TRANSFER VARIATION



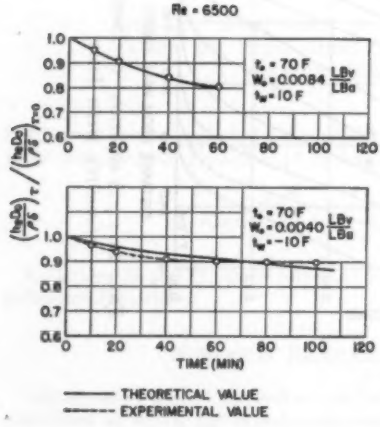


FIG. 27 (*left*). . . .COMPARISON OF THEORETICAL VALUES AND EXPERIMENTAL VALUES OF TOTAL HEAT TRANSFER VARIATION (RE = 6500)

transfers calculated along the laminar region of the cylinder surface were then compared to the corresponding test data. A typical comparison is shown in Fig. 22. The steady state overall heat transfer for the entire cylinder was discussed in Part I.

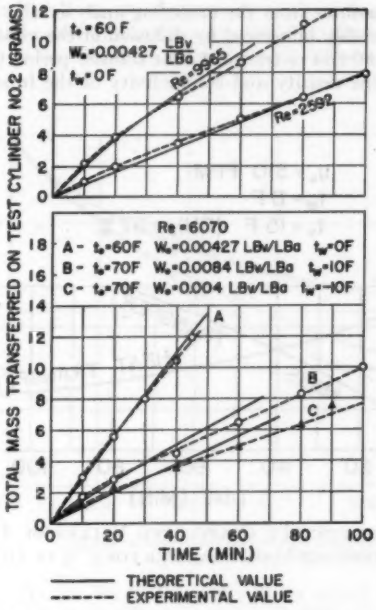

TOTAL MASS TRANSFERRED ON TEST CYLINDER NO.2 (GRAMS)
t_a = 60 F
W_a = 0.00427 LBv/LBa
t_w = 0 F
Re = 3985
Re = 2592
Re = 6070
A - t_a = 60F W_a = 0.00427 LBv/LBa t_w = 0F
B - t_a = 70F W_a = 0.0084 LBv/LBa t_w = 10F
C - t_a = 70F W_a = 0.004 LBv/LBa t_w = -10F
A
B
C
TIME (MIN.)
THEORETICAL VALUE
EXPERIMENTAL VALUE

FIG. 28. . . .COMPARISON OF THEORETICAL VALUES AND EXPERIMENTAL VALUES OF MASS TRANSFER

The finite difference equations corresponding to the diffusion Equations 28 and 31 are solved numerically for the diffusion through a frost layer for the following boundary conditions:

For all τ: $m_x = O$, at $x = O$; $m_x = -\delta(1 - \rho_t/\rho_{to})\,\Delta C/\Delta x$, at $x = L$; $t_x = 0$ F, at $x = O$; $t_x = 20$ F, at $x = L$.
For all x: S. G. = 0.2, for $\tau = O$, $L = \frac{1}{8}$ in.

Figs. 23 and 24 show the results of the calculations.

In Fig. 24 the conductivity variation curve of the frost is subdivided into 3 different periods. There is the first period (A in Fig. 24) during which variation of conductivity is small. Then a rapid increase is observed in the following period (B in Fig. 24). Finally, the conductivity approaches a limiting value during the period C. A similar pattern is also observed in one of the experiments found in Reference 1 (Part I). The absolute length of the periods and magnitude of the increases will

depend on the particular boundary conditions. However, it is thought that the diffusion process of vapor through the frost will result, in general, in the mode of variation shown. The quasi-steady state in heat transfer found in Part I is then reached mainly as the period of the rapid increase in the conductivity (period B) begins.

One may logically assume from the preceding analysis that the conductivity of the frost is not appreciably influenced by diffusion of the vapor through the frost during the first 60 to 80 min of transient heat transfer period (Part I). Then, the time dependency of the density and conductivity of the frost during this period

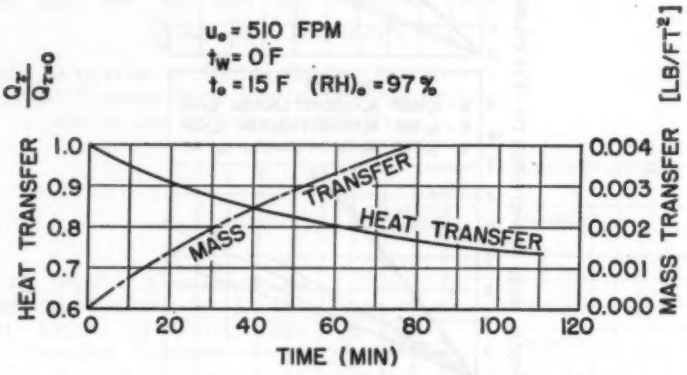


FIG. 29. . . .THEORETICALLY CALCULATED VALUES OF HEAT TRANSFER VARIATION AND MASS TRANSFER FOR A ½ IN. OD TUBE

may be neglected. It is known, however, that the frost formed under different conditions has different properties.

Here the Equations 16, 18 and 20 are solved in conjunction with the experimental results so that the average density and conductivity of the frost formed under the different conditions during the transient period could be obtained.

Values of the densities and the conductivities were chosen from Fig. 21 for given tested conditions. Equations 16, 18 and 20 were then solved, and the heat transfer variation was obtained. The calculated result for the first 60 to 80 min was compared to the corresponding experimental result. The solution and the comparison with the test result were continued until a good correlation was obtained. These trial-and-error type calculations were performed for 3 Reynolds numbers and $(t_o - t_w)/(W_o - W_w)$, the same conditions at which some of the experiments were conducted in Part I. After much study, it was decided to plot the average frost densities thus obtained for the first 60 to 80 min as a function of the total heat transfer potential, $h_e\,(e_o - e_w)$ (see Part I). Fig. 25 shows this result. Figs. 26 and 27 show the correlations between the calculated and tested values of the total heat transfer variation. Fig. 28 shows similar correlations for mass transfer.

It is seen in Figs. 26, 27, and 28 that Equations 16, 18 and 20 approximate the variations of heat and mass transfers well when they are solved with the aid of Fig.

25. Therefore, the equations may be used with Fig. 25 to calculate the heat and mass transfers for the first 60- to 80-min transient periods for different combinations of Re, $(t_o - t_w)/(W_o - W_w)$, and $h_o\,(e_o - e_w)$. After the first 60- to 80-min period, the process becomes quasi-steady. This condition was discussed in Part I.

The experiment in Part I was conducted essentially with one diameter cylinder. Therefore nothing can be said with certainty for different diameter cylinders. However, it is reasonable to assume, from the derivations of the equations used and the

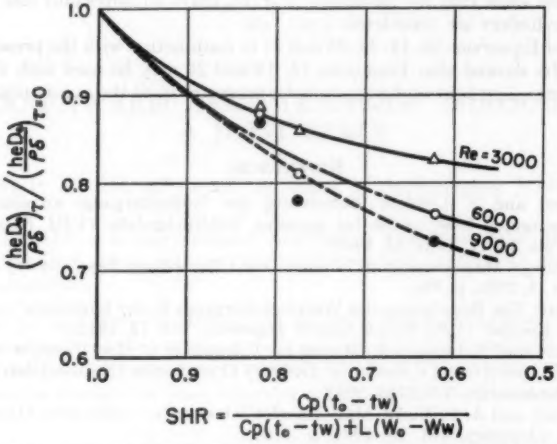


FIG. 30. . . .TOTAL HEAT TRANSFER RATE AFTER EQUILIBRIUM IS ESTABLISHED (TIME >80 MIN) VS SENSIBLE HEAT RATIO

close correlations found between the calculated and experimental values, that the preceding analyses would be applicable to any other diameter cylinders. The variations of heat and mass transfer are calculated by the Equations 16, 18, and 20 and Fig. 25 for a ½-in OD tube with U_o = 510 fpm, t_w = 0 F, t_o = 15 F, and 97 percent relative humidity of the air stream. The result appears in Fig. 29.

CONCLUSIONS

The ratio of Stanton number of mass transfer to that of heat transfer was found from the experiment to be 0.88. This agreed closely with the ratio of 0.90 obtained for a flat plate in Reference 11. The Stanton number ratio was 1.036 when it was calculated by Colburn's analogy between heat and mass transfer.

The theoretically developed Equation 13 for steady state heat transfer along the laminar boundary layer on the cylinder enables one to calculate the local heat transfer with the possible error of 10 percent, and the overall heat transfer from θ = 0 deg to θ = 70 deg with the possible error of 6 percent.

Typical variations of local heat transfers with time are shown on Figs. 10 and 15. The test results showed that a quasi-steady state in total heat transfer was reached in 60 to 80 min for the cases tested (Figs. 11, and 12). A similar phenomenon was also observed in Reference 14, (Fig. 1) for a flat plate.

Fig. 30 plotted from Figs. 7 and 16 shows that the decrease in the heat transfer to the quasi-steady state is largely influenced by the Reynolds number and the sensible heat ratio for a given diameter of a cylinder. The decrease becomes greater as the sensible heat ratio decreases and as the Reynolds number increases. The theoretically developed Equations 16, 18 and 20 also bear out similar facts. The equations also show that the parameter $U_o\tau/D_o$ plays an important role when different size cylinders are considered.

Solution of Equations 16, 18, 20, 28 and 31 in conjunction with the present experimental results showed that Equations 16, 18 and 20 may be used with Fig. 25, to predict the transient heat and mass transfer processes until the quasi-steady state is reached.

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1662

No. 1662

DOWNWARD HEAT TRANSFER THROUGH A JOIST SPACE

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THE EXISTING, or conventional, method[1] of evaluating total heat transfer through a joist space, without insulating materials, starts with the following separate calculations.

1. Conduction and convection through the air space (obtainable by experiments only).
2. Radiation between the upper and the lower plates with the assumption that they are infinite parallel plates (joists are completely ignored in the radiation consideration).
3. Conduction through the joists assuming a linear temperature variation along the surface.

The total heat transfer through the joist space is then obtained by summing up these 3 values. Extensive experiments at the University of Minnesota have shown that often this conventional method of calculation does not yield heat transfer values with adequate accuracy.

In this paper, a theoretical study is made of the mode of heat transfer downward through a typical joist space. The theoretical work is then compared to some of the pertinent experimental data.

CONVECTION IN THE JOIST SPACE

In heat transfer down through a joist space, no natural convection of air is possible when the temperature varies linearly along the joist surface from the upper plate to the lower plate. The mode of heat transfer, then, is by radiation and conduction only.

Usually, however, the radiation heat exchange between the joist surfaces (c in Fig. 1), and the other 2 surfaces (a & b in Fig. 1), causes the temperature variation along the joist to deviate from the linear variation. This creates horizontal tem-

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[1] Exponent numerals refer to References.
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perature differences between the joist surface and the air. The adjacent air is then heated by the joist and rises along its surface. Upon reaching the upper plate, the air stream continues its movement along this plate horizontally toward the middle of the joist space. At midway, the stream meets the counter stream created

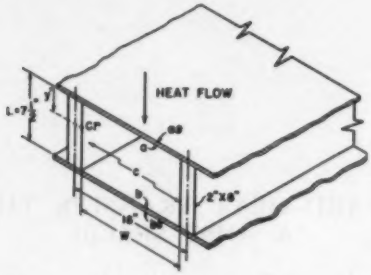

FIG. 1. . . .DIAGRAM OF THE JOIST SPACE STUDIED

by the other joist. The 2 streams merge and descend toward the lower plate. The overall picture is represented in Fig. 2. Fig. 3 shows an enlarged portion of the joist space and the general shape of the velocity and the temperature profile through the boundary layer.

The simplified theoretical model shown on Fig. 4, is assumed here in order to solve the complicated problem shown on Figs. 2 and 3. In Fig. 4, L represents

FIG. 2. . . .REPRESENTATION OF AIR CURRENTS IN THE JOIST SPACE

the joist and $W/2$ represents half of the upper plate. Therefore, in the theoretical study, the effect of turning the boundary layer at the upper corner is discarded. This is justified because since the velocities involved in the boundary layer are small, the turning does not distort the velocity and the temperature profiles to any degree.

The surface temperature of the joist is assumed to be uniform at the average temperature, T_s.

T_s usually rises above T_o due to the radiation heat exchange. T_o represents the temperature of the air outside the boundary layer. The actual magnitude of T_s

depends largely on the radiation itself which, in turn, is a function of the shape factors, surface emissivities, and T_a and T_b.

This excess temperature of T_c above T_o acts as a driving force, and a natural convection boundary layer is built up along the joist surface. After the air stream reaches $x = L$, the upward driving force disappears since the actual direction of the stream is horizontal for $x > L$ (Figs. 2 and 4). However, for $x > L$, the air within the boundary layer still possesses momentum. This momentum was imparted to the air previously while it was traveling along the joist surface. The velocities of the air in the boundary layer decrease continuously for $x > L$ because of the drag

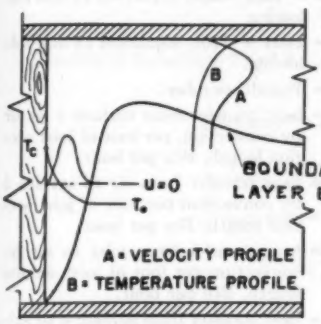

FIG. 3. . . .ENLARGED PORTION OF JOIST SPACE SHOWING GENERAL SHAPE OF VELOCITY AND TEMPERATURE PROFILES THROUGH THE BOUNDARY LAYER

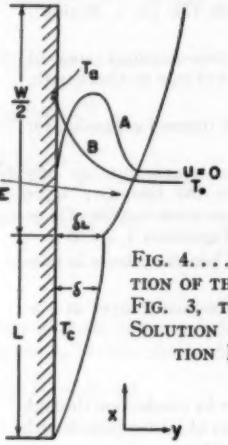

FIG. 4. . . .REPRESENTATION OF THE CURVES OF FIG. 3, TO ASSIST IN SOLUTION OF CONVECTION PROBLEM

at the solid surface and the continuous increase in the thickness of the boundary layer.

The heat transfer between the joist and the air can be obtained from the existing solution of the natural convection boundary layer equations for a vertical heated surface.

Integral Equations 1 and 2 of momentum and thermal boundary layers are derived in Reference 2.

$$\frac{d}{dx}\int_0^{\delta_m} u^2\,dy = g\beta\int_0^{\delta_m} D\,dy - \nu\left(\frac{du}{dy}\right)_0 \qquad (1)$$

$$\text{and} \quad \frac{d}{dx}\int_0^{\delta_t} uD\,dy = -\alpha\left(\frac{dD}{dy}\right)_0 \qquad (2)$$

In the solution of Equations 1 and 2, the temperature and velocity profiles are usually approximated by the parabolas of Equations 3 and 4.

$$D = \theta\left(1 - \frac{y}{\delta}\right)^2 \qquad (3)$$

NOMENCLATURE

A_a = inside surface area of upper plate per foot of joist section length, square feet.

A_b = inside surface area of lower plate per foot of joist section length, square feet.

A_c = surface area of joist per foot of joist section length (2L for a section), square feet.

A_w = horizontal cross-sectional area of joist, per foot of joist section length, square feet.

β = coefficient of thermal expansion for air.

D = difference in temperature at any point within the boundary layer and the temperature outside of it = $(T - T_o)$, Equations 1, 2 and 3.

δ = thickness of boundary layer in general, feet.

δ_L = thicknes of boundary layer at $x = L$, feet.

$\Delta t = T_a - T_b$.

e = emissivity.

E = heat transfer by conduction through joist, per foot of joist section length Btu per (hour).

F = radiation shape factor.

g = 32.2 feet per (second) (second).

h_a = local convective heat transfer coefficient for surface a, Btu per (hour) (square foot) (Fahrenheit degree).

h_b = local convective heat transfer coefficient for surface b, Btu per (hour) (square foot) (Fahrenheit degree).

H_c = average convective heat transfer coefficient for surface c, Btu per (hour) (square foot) (Fahrenheit degree).

K = thermal conductivity of air, Btu per (hour) (foot) (Fahrenheit degree).

K_w = thermal conductivity of joist material, Btu per (hour) (foot) (Fahrenheit degree).

α = thermal diffusivity of air.

l_m = thickness of momentum boundary layer, feet.

l_t = thickness of thermal boundary layer, feet.

L = depth of the joist space defined in Fig. 1, feet.

ν = kinematic viscosity of air, square feet per hour.

$n_1 = -105\nu - 60\alpha$, Equation 13 and following.

$n_2 = 240\alpha + 210\nu$, Equation 15 and following.

Pr = Prandtl number.

q_a = heat transfer from surface a to air by convection, per foot of joist section length, Btu per hour.

q_b = heat transfer from air to surface b by convection per foot of joist section length, Btu per hour.

q_c = heat transfer from joist to air by convection per foot of joist section length, Btu per hour.

q_{ab} = heat transfer from surface a to surface b by radiation per foot of joist section length, Btu per hour.

q_{ac} = heat transfer from surface a to surface c by radiation per foot of joist section length, Btu per hour.

q_{cb} = heat transfer from surface c to surface b by radiation per foot of joist section length, Btu per hour.

T = Rankine temperature unless specified otherwise.

T_a = surface temperature of the upper plate, Rankine degrees.

T_b = surface temperature of the lower plate, Rankine degrees.

T_c = average surface temperature of joist used in convection analysis, Rankine degrees.

T_o = temperature of air outside boundary layer, Rankine degrees.

$\theta = T_o - T_o$.

u = velocity in direction parallel to solid surface, feet per hour.

U_1 = a function of x defined in Equation 4.

W = width of the joist space defined in Fig. 1, feet.

x,y = directions and distances defined in Fig. 4.

y' = distance defined in Fig. 1, feet.

$Z = \delta^3$, Equation 12 and following, square feet.

Subscripts

a, b, c = surfaces defined in Fig. 1.

ab, ac, cb = a to b, a to c, c to b, respectively.

ao, bo = surfaces defined in Fig. 1.

cr = a point on joist surface at a distance 2/3 L down from upper plate.

L = (x = L)

o = condition at the solid boundary, (y = O).

$$u = U_1 \frac{y}{\delta}\left(1-\frac{y}{\delta}\right)^2, \text{ where } \delta = l_m = l_t^* \text{ ------(4)}$$

Solution of Equations 1 and 2 yields Equations 5 and 6.

$$\delta = 3.93 Pr^{-1/5}(0.952+Pr)^{1/4}\left(\frac{g\beta\theta}{\nu^2}\right)^{-1/4} x^{1/4} \text{ ------(5)}$$

$$U_1 = 5.17\nu\left(\frac{20}{21}+Pr\right)^{-1/2}\left(\frac{g\beta\theta}{\nu^2}\right)^{1/2} x^{1/2} \text{ ------(6)}$$

It was shown in reference (2) that

$$H_c = \frac{4}{3}\left(\frac{2K}{\delta_L}\right) \text{ ------(7)}$$

Detailed derivations of Equations 1 through 7 are given in Reference 2.

The heat transfer from the joist to the air, q_c, is then obtained by Equation 8.

$$q_c = H_c A_c (T_c - T_o) = H_c A_c \theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The solution of heat transfer for $x > L$ is attempted here using the following procedure.

The momentum equation in the absence of the driving force becomes:

$$\frac{d}{dx}\int_0^{l_m} u^2 dy = -\nu\left(\frac{du}{dy}\right)_0 \text{ ------(9)}$$

The thermal boundary layer equation is unchanged from Equation 2.

A substitution of Equations 3 and 4 into Equations 9 and 2 yields Equations 10 and 11.

$$1/105 d/dX(U_1^2\delta) = -\nu U_1/\delta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

$$1/30 d/dX(U_1\delta) = 2\alpha/\delta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Equations 10 and 11 are combined to yield Equation 12.

$$\delta^3\, dU_1/dX = -105\nu - 60\alpha. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Let

$$Z = \delta^3$$

---

* $l_m = l_t$ is a popular accepted assumption for the natural convection analysis.

and

$$n_1 = -105\nu - 60\alpha$$

Then Equation 12 becomes:

$$Z\,dU_1/dX = n_1 \quad \ldots\ldots\ldots\ldots \quad (13)$$

Equation 11 may be transformed to the following equation:

$$\delta^2 \frac{dU_1}{dx} + \tfrac{1}{2} U_1 \frac{d\delta^2}{dx} = 60\alpha.$$ *With the aid of Equation 13, this becomes:*

$$n_1 + \tfrac{1}{2} U_1 \frac{dZ}{dx} = 60\alpha \quad \ldots\ldots\ldots\ldots (14)$$

Let $n_2 = 240\alpha + 210\nu$, then Equation 14 becomes:

$$U_1\,dZ/dX = n_2 \quad \ldots\ldots\ldots\ldots \quad (15)$$

Now the problem is to solve the 2 Equations 13 and 15 simultaneously. Dividing Equation 15 by Equation 13 yields:

$$dZ/Z = (n_2/n_1)(dU_1/U_1)$$

This is integrated between the limits of $x = L$ to any other $x > L$ as follows:

$$\int_{Z_L}^{Z} \frac{dZ}{Z} = \frac{n_2}{n_1} \int_{U_{1L}}^{U_1} \frac{dU_1}{U_1}, \quad \text{and} \quad Z = Z_L \left(\frac{U_1}{U_{1L}}\right)^{\frac{n_2}{n_1}} \quad \ldots\ldots (16)$$

Differentiate Equation 16 with respect to x, and make a substitution from Equation 15.

$$\text{Then: } (U_1)^{\frac{n_2}{n_1}} dU_1 = \frac{n_1}{Z_L} (U_{1L})^{\frac{n_2}{n_1}} dx \quad \ldots\ldots\ldots\ldots (17)$$

This again is integrated between the limits of $x = L$ to any other $x > L$.

$$\text{Then: } (U_1) = \left[(U_{1L})^{\frac{n_2}{n_1}} \left(\frac{n_1 + n_2}{Z_L} x' + U_{1L}\right)\right]^{\frac{n_1}{n_1 + n_2}} \quad \ldots\ldots (18)$$

where: $x' = x - L$. *Finally, from Equation 16:*

$$\delta = \delta_L \left[\frac{n_1 + n_2}{Z_L U_{1L}} x' + 1\right]^{\frac{n_2}{2(n_1 + n_2)}} \quad \ldots\ldots\ldots\ldots (19)$$

δ_L and U_{1L} can be obtained from Equations 5 and 6 by putting $x = L$. Equation 19 then gives the thickness of the boundary layer, δ, for any x^1. The local heat transfer coefficient for the upper plate, h_a, from the boundary layer solution is $2K/\delta$. Therefore h_a can be obtained from Equation 19.

The total heat transfer from the upper plate is finally obtained by the following integration:

$$q_a = 2(T_a - T_s) \int_0^{\frac{W}{2}} h_a\,dx' \quad \ldots\ldots\ldots\ldots (20)$$

The heat transfer coefficient between the lower plate and the air cannot be obtained so easily because the behavior of the air streams after merging is unknown. After studying the physical conditions involved, it was decided to assume that $h_b = h_a$ for the present analysis. The close agreement of the calculated values to the experimental results, seen later, would also justify this assumption.

RADIATION AND CONDUCTION THROUGH THE JOIST

As stated earlier in this paper the conventional way of evaluating the radiation is to consider that the plates a and b are infinite in size to neglect the effect of the joists on radiation completely. It was stated[1] that this assumption did not introduce any sizable error. This is true if it is also assured that the temperature variation through the joists is unaffected by radiation and is always linear.

Experimental results at the University of Minnesota have shown that this assumption may lead to considerable discrepancy in the evaluation of the total conductance of the joist space. This is because the temperature gradient through the joist affects the conduction through the joists themselves and the convection as well as the radiation between the joist and the surfaces a and b.

Hottell[3] gave Equation 21 for the radiation shape factor between 2 plates such as a and b shown in Fig. 1.

$$F_{ab} = \frac{L}{W} + \sqrt{1 + \left(\frac{L}{W}\right)^2} \qquad (21)$$

Then, by the reciprocity theorem; $F_{ba} = F_{ab}$; $F_{ac} = F_{bc} = 1 - F_{ab}$; $F_{ca} = F_{cb} = A_a/A_c\, F_{ac}$; and $F_{cc} = 1 - 2F_{ca}$.

Rational radiation equations for the net heat transfer between the surfaces a and b, a and c, and c and b, are as follows:†

$$q_{ab} = 0.173\, e_a e_b A_a S_2 S_3 S_4 \left[F_{ab} + (1-e_c) F_{ac} F_{cb} S_1\right] \left[\left(\frac{T_a}{100}\right)^4 - \left(\frac{T_b}{100}\right)^4\right] \qquad (22)$$

where: $S_1 = \dfrac{1}{1 - F_{cc}(1-e_c)}$; and

$$S_2 = \frac{1}{1 - (1-e_a)(1-e_c) F_{ac} F_{ca} S_1}; \text{ and}$$

$$S_3 = \frac{1}{1 - (1-e_b)(1-e_c) F_{bc} F_{cb} S_1}; \text{ and}$$

$$S_4 = 1 \Big/ \Big\{1 - (1-e_a)(1-e_b) S_2 S_3 \left[F_{ba} + (1-e_c) F_{bc} F_{ca} S_1\right] \left[F_{ab} + (1-e_c) F_{ac} F_{cb} S_1\right]\Big\}$$

$$q_{ac} = 0.173\, e_a e_c A_a S_2' S_3' S_4' \left[F_{ac} + (1-e_b) F_{ab} F_{bc}\right] \left[\left(\frac{T_a}{100}\right)^4 - \left(\frac{T_{cr}}{100}\right)^4\right] \qquad (23)$$

† See Reference 3 for the detailed derivation of the radiation equations.

$$\text{where: } S_2' = \frac{1}{1-(1-e_a)(1-e_b)F_{ab}F_{ba}}; \text{ and}$$

$$S_3' = \frac{1}{1-(1-e_c)(1-e_b)F_{cb}F_{bc}}; \text{ and}$$

$$S_4' = 1/\left\{1-(1-e_a)(1-e_c)S_2'S_3'\right.$$
$$[F_{ca}+(1-e_b)F_{cb}F_{ba}]$$
$$\left.[F_{ac}+(1-e_b)F_{ab}F_{bc}]\right\}$$

$$q_{cb} = 0.173 e_c e_b A_a S_2'' S_3'' S_4'' [F_{bc}+(1-e_a)F_{ba}F_{ac}]$$
$$\left[\left(\frac{T_{cr}}{100}\right)^4 - \left(\frac{T_b}{100}\right)^4\right] \quad \text{---------- (24)}$$

$$\text{where: } S_2'' = \frac{1}{1-(1-e_b)(1-e_a)F_{ba}F_{ab}}; \text{ and}$$

$$S_3'' = \frac{1}{1-(1-e_c)(1-e_a)F_{ca}F_{ac}}; \text{ and}$$

$$S_4'' = 1/\left\{1-(1-e_b)(1-e_c)S_2''S_3''\right.$$
$$[F_{cb}+(1-e_a)F_{ca}F_{ab}]$$
$$\left.[F_{bc}+(1-e_a)F_{ba}F_{ac}]\right\}$$

The temperature and the shape factors along the joist surface vary non-linearly from the upper plate to the lower plate. The radiation Equations 22 through 24 were developed for the representative surface temperature T_{cr} and with the overall shape factors between the joist and the surfaces a and b. The question here is: "Which point along the joist surface does T_{cr} represent?"

Let y' indicate the distance along the joist surface from the upper plate to the point where the temperature is at T_{cr} (see Fig. 1). Then, in the heat balance (next section), the following simple equations will be used to calculate the conduction through the joist.

$$E_{ac} = K_w A_w (T_a - T_{cr})/y' \quad \text{. . . . . (25)}$$

$$E_{cb} = K_w A_w (T_{cr} - T_b)/(L - y') \quad \text{. . . . . (26)}$$

It is known from the experiments that usually the temperature varies along the joist surface in the general manner shown in Fig. 5. Through much analysis with the equations previously developed, it was found that the best approximation to the actual temperature variation along the joist surface is obtained by letting $y' = 2/3\ L$.

The heat transfer from the upper plate to the lower plate by the conduction through the joist is actually (See Fig. 5) given by Equation 27.

$$E_{ac}\ (\text{Actual}) = K_w A_w \left(\frac{dt}{dy}\right)_1 \quad \text{---------- (27)}$$

whereas that calculated by the conventional method is as stated by Equation 28.

$$E_{ab}\left(\begin{matrix}\text{Conventional}\\ \text{method}\end{matrix}\right)=K_w A_w\left(\frac{dt}{dy}\right)_2=K_w A_w\frac{T_a-T_b}{L}\quad\text{....(28)}$$

and that calculated by the present method is stated by Equation 29.

$$E_{ab}\left(\begin{matrix}\text{Present}\\ \text{method}\end{matrix}\right)=K_w A_w\left(\frac{dt}{dy}\right)_3=K_w A_w 3\frac{T_{cr}-T_b}{L}\quad\text{...(29)}$$

Comparison of the equations in light of Fig. 5 illustrates the improvement of the present method over the conventional one in the calculation of the conduction through the joist.

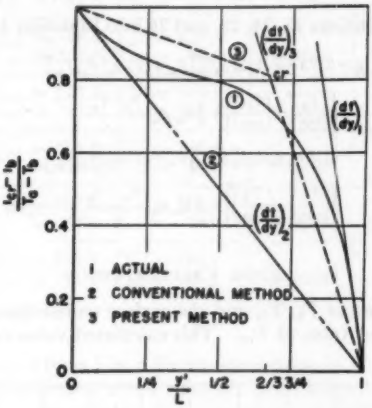

FIG. 5. . . .TYPICAL TEMPERATURE DISTRIBUTION ALONG THE JOIST SURFACE

HEAT BALANCES

The heat transfer between the air in the joist space and the surrounding surfaces is by convection only. Therefore the heat balance of Equation 30 can be set up for the air.

$$q_c + q_a = q_b \quad\text{. . . . . (30)}$$

With the assumption of $h_b = h_a$ explained in the section on *convection*, Equation 30 becomes Equation 31.

$$(T_c - T_b)/(T_a - T_c) = 1 + q_c/q_a \quad\text{. . . . . (31)}$$

Substitution of Equations 8 and 20 into Equation 31 yields Equation 32.

$$\frac{T_c-T_b}{T_a-T_c}=1+\frac{H_c A_c(T_c-T_a)}{2(T_a-T_s)\int_0^{y_b} h_a dx'}\quad\text{.....(32)}$$

The average temperature of the joist surface, T_o, is obtained from the theoretical temperature variation curve (curve 3 in Fig. 5) for a given value of T_{cr}. Equation 32 can be solved for T_o as a function of T_{cr} for the given values of T_a and T_b. With the functional relation between T_o and T_{cr} known, Equations 8 and 20 can be solved for q_c and q_a for the given values of T_a, T_b and T_{cr}.

For the limited range of the temperatures encountered here, it was possible to express the calculated values of q_c and q_a as the functions of $(T_{cr} - T_b)/(T_a - T_b)$ and $T_a - T_b = \Delta t$. This is true because a convection heat transfer is essentially based on the temperature differences rather than the absolute temperatures. Fig. 6 shows these values of q_c and q_a calculated.

Next, a heat balance is set up on the joist in accordance with Equation 33.

$$q_o = q_{ao} + E_{ao} - q_{ob} - E_{ob} \quad \ldots\ldots\ldots \quad (33)$$

Substitution of Equations 23, 24, 25, and 26 into Equation 33 yields Equation 34.

$$q_c = 0.173 e_a e_c A_a S_2' S_3' S_4' \left[F_{ac} + (1-e_b) F_{ab} F_{bc}\right] \left[\left(\frac{T_a}{100}\right)^4 - \left(\frac{T_{cr}}{100}\right)^4\right] + \frac{3}{2} K_w A_w \frac{T_a - T_{cr}}{L}$$
$$- 0.173 e_c e_b A_a S_2'' S_3'' S_4'' \left[F_{bc} + (1-e_a) F_{ba} F_{ac}\right] \left[\left(\frac{T_{cr}}{100}\right)^4 - \left(\frac{T_b}{100}\right)^4\right] - 3 K_w A_w \frac{T_{cr} - T_b}{L} \quad \text{-------(34)}$$

FINAL CALCULATIONS

For the given values of T_a, T_b, and the surface emissivities, Equation 34 is first solved for an assumed value of T_{cr}. This calculated value of q_c is then compared

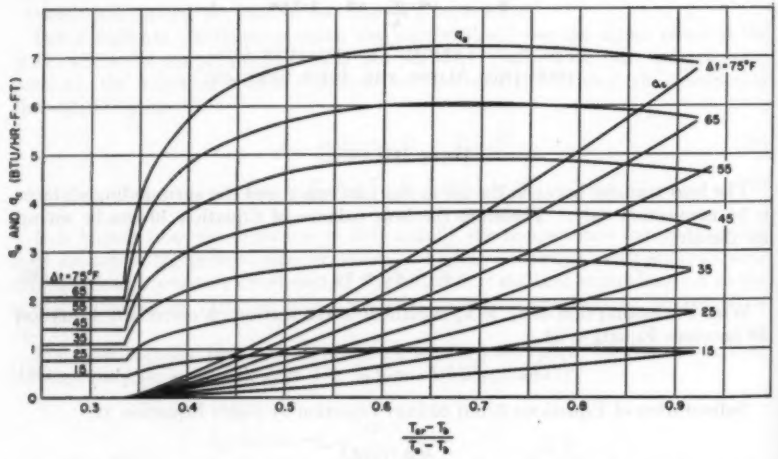

FIG. 6. . . .VARIATIONS OF q_a AND q_c FOR A JOIST SPACE 16-IN. WIDE X 7½-IN. HIGH AT VARIOUS TEMPERATURE DIFFERENCES

TABLE 1. . . . COMPARISON OF THE CONDUCTANCES CALCULATED BY THE PRESENT METHOD TO THOSE OBTAINED BY EXPERIMENTS AND TO THOSE CALCULATED BY THE CONVENTIONAL METHOD

SURFACE CONDITIONS	TEST CONDITIONS, TEMPERATURES, FAHRENHEIT	CONDUCTANCE BASED ON T_{ao} AND T_{bo} BTU PER (HR) (SQ FT) (F DEG)		
		EXPERIMENTAL RESULTS	CALCULATED RESULTS	
			BY CONVENTIONAL METHOD	BY PRESENT METHOD
Bare surfaces	$T_{ao} = 122.2$; $T_a = 109.0$ $T_b = 97.8$; $T_{bo} = 87.9$	0.3900	0.4227	0.3898
	$T_{ao} = 48.4$; $T_a = 127.8$ $T_b = 103.8$; $T_{bo} = 95.8$	0.4070	0.4417	0.4080
Aluminum foil on b	$T_{ao} = 126.3$; $T_a = 119.9$ $T_b = 83.6$; $T_{bo} = 80.7$	0.1370	0.0922	0.1365
	$T_{ao} = 154.3$; $T_a = 143.9$ $T_b = 89.4$; $T_{bo} = 84.6$	0.1460	0.1060	0.1500
Aluminum foil on b and c	$T_{ao} = 127.4$; $T_a = 122.8$ $T_b = 82.4$; $T_{bo} = 79.9$	0.0983	0.0994	0.0956
	$T_{ao} = 155.9$; $T_a = 148.4$ $T_b = 87.4$; $T_{bo} = 83.2$	0.1060	0.1172	0.1038

to the value of q_c obtained from Fig. 6 for the same value of T_{cr}. If these 2 values of q_c do not agree with each other, a new value of T_{cr} is assumed, and the calculation of Equation 34 and the comparison to Fig. 6 are repeated until an adequate agreement is reached. This solution by the method of iteration yields the definite values of q_c and T_{cr} desired. q_a is read off Fig. 6 directly. q_{eb} is calculated from Equation 24.

Finally, the overall heat transfer through a section of joist is obtained from Equation 35.

$$q_{Total} = q_{ab} + q_{eb} + E_{eb} + q_b \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

Three sets of the overall heat transfer rates per unit temperature difference, $q_{Total}/(T_a - T_b)$, are calculated and are plotted in Figs. 7, 8, and 9. They are for the cases with bare inside surfaces, with aluminum foil on the surface b, and with foil on the surfaces b and c, respectively. The heat transfer rates are calculated and shown on the figures for a section of the 16-in. wide, 7½-in. deep and 1-ft long joist (see Fig. 1).

The emissivities used are 0.9 and 0.03 for the bare surface and the aluminum foil respectively.

COMPARISON WITH EXPERIMENTAL DATA

Six sets of experimental data are used here for the sake of comparison and support to the present theoretical elaborations. These sets are taken from the extensive

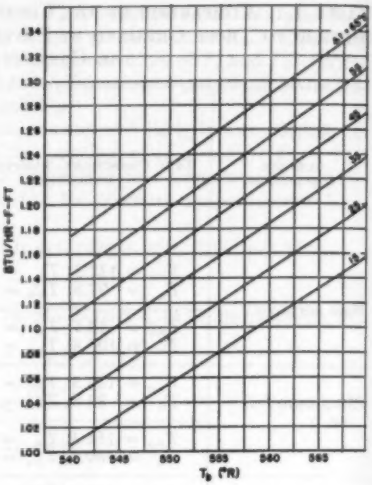

FIG. 7. . . .RATE OF TOTAL HEAT TRANSFER FOR 1-FT OF JOIST SECTION 16-IN. WIDE X 7½-IN. HIGH X 1-FT LONG, IN BTU PER (HOUR) (FOOT) (FAHRENHEIT DEGREE) WITH BARE INSIDE SURFACES, $e_a = e_b = e_c = 0.90$

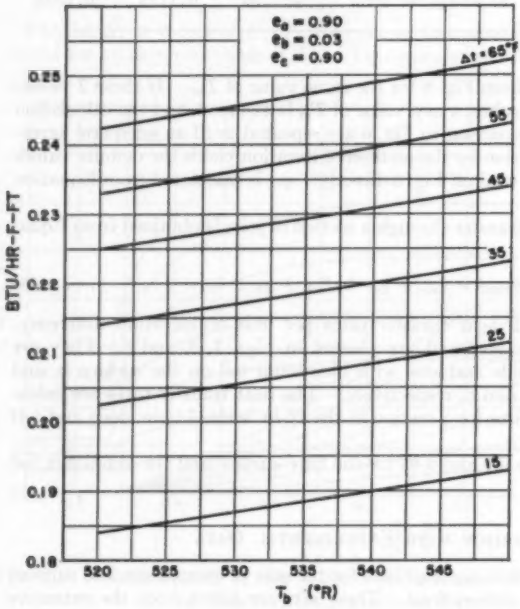

FIG. 8. . . .RATE OF TOTAL HEAT TRANSFER FOR 1-FT OF JOIST SECTION 16-IN. WIDE X 7½-IN. HIGH X 1-FT LONG, IN BTU PER (HOUR) (FOOT) (FAHRENHEIT DEGREE) WITH ALUMINUM FOIL ON BOTTOM SURFACE OF JOIST SPACE

experimental data presented in Reference 4. The particular sets are chosen because they were obtained through the experiments with the 3 cases of the joist spaces considered in Figs. 7, 8, and 9.

In the experiment, the plate a was made of ¼-in. plywood and the plate b was made of ½-in. gypsum board. The conductances of the plates were 1.033 Btu per (hr) (sq ft) (F deg) and 2.80 Btu per (hr) (sq ft) (F deg) respectively. The tem-

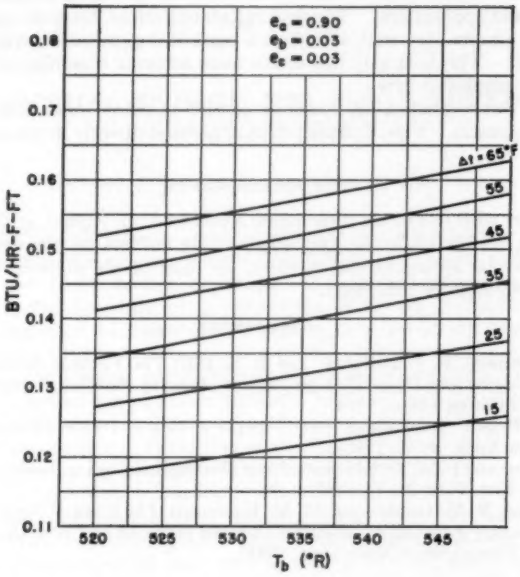

FIG. 9. . . .RATE OF TOTAL HEAT TRANSFER FOR 1-FT OF JOIST SECTION 16-IN. WIDE X 7½-IN. HIGH X 1-FT LONG, IN BTU PER (HOUR) (FOOT) (FAHRENHEIT DEGREE) WITH ALUMINUM FOIL ON BOTH BOTTOM AND JOIST SURFACES OF JOIST SPACE

peratures T_{ao}, T_a, T_{bo}, T_b, (see Fig. 1) and the temperatures along the joist surface were measured as well as the heat transfer rates during the tests. Reference 4 describes the test in detail.

The total heat transfer rates are calculated from Figs. 7, 8, and 9, for the measured values of T_a and T_b. The overall conductance based on T_{ao} and T_{bo} were then calculated with the known conductances of the plates a and b. These calculated values are compared to the measured values of the conductances in Table 1.

The conductances are also calculated by the conventional method and are presented in the same table. Agreement of the new calculated values with the experi-

mental results is strikingly good. The improvement of the present method over the conventional method is also shown.

CONCLUSIONS

The analyses shown here present a better insight into the mechanisms of the complicated downward heat transfer through a typical joist space.

A substantial number of simplifying assumptions were made during the course of the theoretical derivations in order to make the analyses more suitable to practical calculations and applications. The final calculated values, however, agree with the experimental results very well, and show a marked improvement over the conventional method. This is largely due to the more accurate reasoning of the mode of heat transfer employed here.

The basic theory studied here may be applied to many other joist section geometries and surfaces. Figs. 7, 8, and 9, may be used directly in the application.

ACKNOWLEDGMENTS

The authors wish to express their appreciation to A. B. Algren, at the University of Minnesota, for his invaluable suggestions. The authors are also deeply indebted to Evert Lehtola, an engineering student, for most of the numerical calculations which he performed for this paper.

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DISCUSSION

H. C. BROWN, Lancaster, Pa. (WRITTEN): The authors are to be commended for this work which will be of invaluable help to those desiring accurate calculations of downward heat transfer through a typical joist space. It is gratifying to note the good agreement between the present method of calculation and experimental results. The discrepancy in the old conventional method of calculation is ample evidence that it is truly out of date.

1663

No. 1663

ANNUAL ENERGY USE FOR ELECTRIC SPACE HEATING

By R. L. Boyd†, Pittsburgh, Pa.

THE IMPRESSIVE increase in use of electricity for space heating of residences, as well as commercial and industrial establishments, merits close analysis of this type of energy by heating engineers. This paper presents typical data on electric heat installations and brings out definite differences in quantities of energy required for operation as compared to estimates for conventional heating systems.

Despite the fact that upwards of half a million residences now use electricity as their sole source of heating, authentic data on the operating characteristics of electric heating systems have not been generally available. The examples presented here emphasize the need for study of such characteristics with adequate instrumentation to permit proper evaluation of the data and establishment of standards for estimating the energy consumption of electric heating systems.

Evaluation of combustion heating systems has been largely devoted to determination of useful heat delivered to the space compared to the heat value of the fuel input. With resistance electric heat, determination of such efficiency is unnecessary as conversion of electricity to heat is at 100 percent efficiency and heaters are most frequently located within the spaces being heated. As with other heating means, the selection of type of electric system should be based upon: (*a*) choice of appearance; and (*b*) evaluation of such characteristics as air movement, proportion of radiant and convected heat, visible light emissivity, quietness of operation and operating temperatures.

In Fig. 1, a plot of fuel costs in principal cites of the United States[1] shows a wide range of costs per million Btu of delivered heat for various conventional fuels, and for electricity at unit costs per kwhr well below usual residential electric rates. The efficiencies used for combustion fuels are stated in the source referenced. The costs for electric heat are based on 100 percent efficiency. While this chart indicates that electricity at ¾ cent per kwhr is not competitive with fuels generally, experience records, and the growth of electric heating, have demonstrated that it is competitive, even at 1¾ cents per kwhr. Because of the wide variation in fuel and

† Product Manager, Commercial Comfort Heating Division, Edwin L. Wiegand Co. Member of ASHAE.
[1] Exponent numerals refer to References.
Presented at the 65th Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Philadelphia, January 1959.

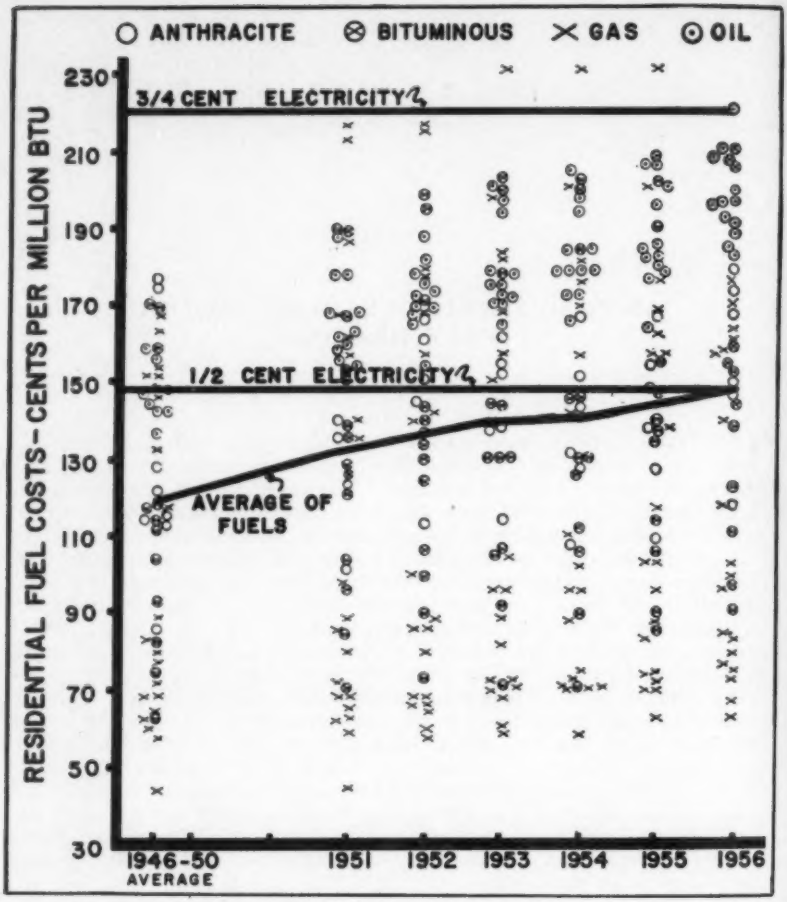

FIG. 1....PLOT OF FUEL COSTS IN PRINCIPAL U.S. CITIES IN RECENT YEARS, WITH AVERAGE AND 2 STATED ELECTRIC RATES SUPER-IMPOSED

electric energy prices due to local conditions, this paper discusses energy requirements since they may be converted to costs by application of local prices.

The conventional methods of estimating fuel consumption have been developed based on use of combustion fuels. Over the years, these methods have been refined so that they yield close estimates, generally, when used with the fuels on which they were based. They are highly inaccurate when used for electric heat intelligently applied, although they are not so far in error when electric heat is applied

to uninsulated or poorly insulated structures. When electric heat is applied on a room-by-room basis to structures properly insulated for an economic balance between first cost and operating cost, conventional estimating methods indicate much more energy required for heating than is shown by available consumption records. This is perhaps partly due to disregard of the internal heat and sun effect which carries a higher proportion of the heating load in the better insulated building, regardless of the heat source used. The influence of internal heat and sun effect

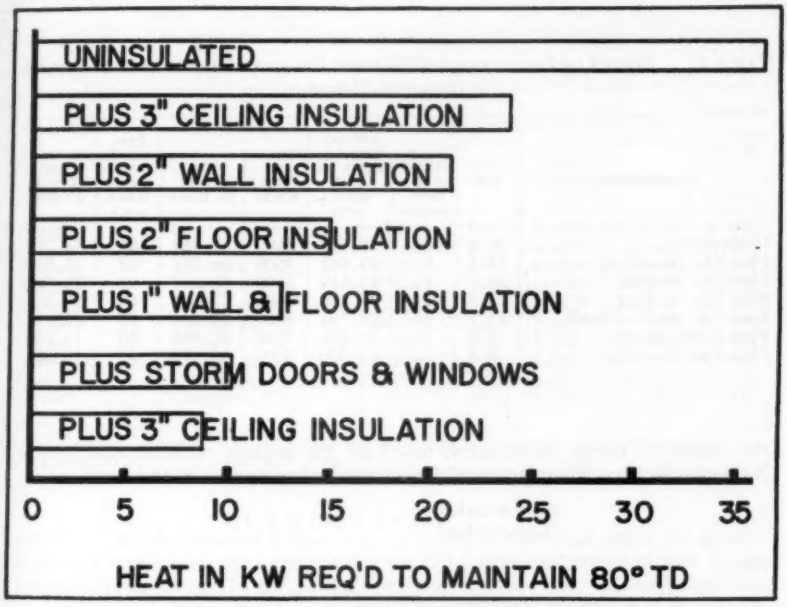

FIG. 2. . . . ILLUSTRATION OF HOW ADDITION OF SUCCESSIVE INSULATIONS AFFECTS ELECTRIC ENERGY REQUIREMENT (BASED ON A 3-BEDROOM HOUSE)

is evidently more significant when a number of thermostats control the heat in each room than with a system regulated from only one central point.

Although the length and severity of the heating season varies with the locality, analysis based on one particular area may illustrate the internal heat influence. As an example, assume a typical 1000 sq ft floor area, brick veneer residence with 8 ft, 3-in. ceiling height, 200 sq ft combined glass and door area, 872 sq ft net wall area and well-vented attic and crawl spaces. The calculated effect of insulation on the capacity required to heat such a residence is shown graphically in Fig. 2. Assume further that 400 kwhr per month of electrical use represents uses releasing heat within the residence and contributing to desired heating 9 months of the year. Based on Fort Wayne, Ind., with 80 deg temperature difference (−10 F design)

and 6,287 degree days, and using the calculated heat loss method in THE GUIDE* to determine the annual requirement for heating, the first 7 columns of Table 1 are developed for the progressive stages of insulation indicated (all steps of insulation are in addition to all preceding steps of insulation).

Comparison of the net kwhr required to heat the residence with the gross calculated kwhr indicates that the internal load contributes from 5 to 19 percent of the calculated heat loss. This has been indicated in previous studies.[2] But actual experience records show that an even more significant deviation from calculated energy requirement is experienced. For example, the last column of Table 1 gives

TABLE 1. . . .EFFECT OF INSULATION ON HEATING CAPACITY REQUIRED IN A TYPICAL 3-BEDROOM HOUSE

CONSTRUCTION	LOAD, KW	HEATING % OF UNINSULATED	ANNUAL KWHR TO HEAT BY CALC. HEAT LOSS	INTERNAL HEAT, KWHR	NET KWHR TO HEAT	PERCENT OF CALC. HEAT LOSS	*NEMA* MANUAL ENERGY
Uninsulated	36.6	100	75,419	3600	71,819	95	53,212
Plus 3 in. on ceiling	24.1	66	49,661	3600	46,061	93	35,038
Plus 2 in. on wall	20.6	56	42,449	3600	38,849	92	29,950
Plus 2 in. on floor	14.9	41	30,704	3600	27,104	88	21,663
Plus 1 in. on flr. & wall	13.7	38	28,231	3600	24,631	87	19,918
Plus storm glazing	10.0	28	20,606	3600	17,006	83	14,539
Plus 3 in. on ceiling	9.4	26	19,370	3600	15,770	81	13,666

the calculated energy requirement based on the formula recommended by the National Electrical Manufacturers' Association[3] using a value of 18.5 for C.

$$\left.\begin{matrix}\text{Annual kwhr} \\ \text{Consumption}\end{matrix}\right\} = HL \times DD \times C/TD$$

where

C = a constant (variable 12 to 24).
HL = heat loss of building, in kw.
DD = annual degree days for the locality.
TD = design temperature difference.

The constant used in the NEMA formula was developed from experience records as were the constants used generally for combustion heating calculations, and even the 18.5 constant (which indicates that about 29 percent of the calculated heat loss is contributed by other sources than the heating system) has given conservatively high energy requirements generally. Experience shows that the formulas developed for predicting fuel consumption of combustion heating systems are not applicable to electric heating without excessive error.[4]

Properly instrumented examples of actual experience records for electrically-heated spaces are difficult to find. Those available are frequently attacked as

* HEATING VENTILATING AIR CONDITIONING GUIDE 1958, p. 467 (published by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, New York, N. Y.).

having been developed by groups vitally concerned that experiences cited will be favorable to electric heat. The experience records referred to herein are actual meter records, but lack desirable detail regarding conditions of operation and sponsorship of a duly recognized Society or committee of such a group. The Appendix, Table A-1, gives *monthly* electrical energy consumption of 10 residences which Table 2 summarizes, illustrating typical results of electric heating installations.

Fig. 3 shows graphically the information in the last 4 columns of Table 2. Listing of many times the examples given would be no more conclusive. The examples

TABLE 2. . . .ELECTRICAL ENERGY CONSUMPTION OF ELECTRICALLY HEATED RESIDENCES

IDENTIFICATION	LOCALITY INDIANA	FLOOR AREA	KILOWATT-HOURS CONSUMED			ESTIMATE IN GENERAL USE IN LOCALITY, KWHR
			TOTAL	NON-HEATING	HEATING	
Cotes	Ossian	1076	17,010	5,580[a]	11,430	9,325
Cohran	Bluffton	1200	19,238	9,864[a]	9,374	10,400
Fechter	Bluffton	1120	13,856	4,608[a]	9,148	9,707
Heline	Ossian	1300	11,740	720[a]	11,020	11,267
Moper	Ft. Wayne	1200	15,718	4,992[a]	10,726	10,400
Shoyer	Ft. Wayne	1400	15,722	6,372[a]	9,350	12,133
Gohert	Ft. Wayne	1100	24,796	13,932	10,864	9,634
Hoard	Ft. Wayne	1500	22,460	8,344	14,116	13,000
Gee[c]	Columbia City	904	16,730	7,849	8,881	7,835
Dufey[c]	Columbia City	904	14,400	6,594	7,806	7,835

[a] Non-heating use estimated by multiplying sum of July and August energy consumption by 6.
[b] Estimated, based on 8¾ kwhr per sq ft floor area.
[c] Identical houses, next door neighbors. Dufey, couple only; Gee, couple and 1-year-old baby.

tabulated include instances with energy consumption over and under the general experience of the area, with the last 4 selected for actual sub-metered data on the heating consumption. In comparing individual cases, it may be concluded that personal living habits influence costs more than weather deviations or assumptions made in setting up the estimates.

Table A-2 in the Appendix gives a statistical record of average total electrical energy for all uses in the electrically heated homes of one utility for 3 years. Through the first year, the same houses were analyzed; in subsequent years, the average of all homes for each month was analyzed. This demonstrates the month by month increase in electrically heated homes on this utility's lines.

In commercial installations, experience records demonstrate actual energy required to be far below net requirements calculated for other fuels. Such installations normally have proportionally greater internal heat sources than residences. In one school built to use electric heat, calculated oil consumption, if oil had been used, ranged from 14,689 gal to 16,015 gal; converting the anticipated net heat delivered to equivalent kwhr of electricity would indicate from 424,789 to 463,135 kwhr of electricity to be required. The first year of operation required 133,480 kwhr to heat the school. The second year was complicated by construction of an additional 4 classrooms, multi-purpose room and kitchen, more than doubling the

electrical connected load. The period from September 1956 through January 1957 was relatively free from influence of the addition, with 69,360 kwhr required for heating, representing 55.8 percent of the normal heating season degree days. Projecting this partial operation to the equivalent of a full season would indicate about 125,000 kwhr to be required for heating this school. This application, running less than one-third the energy requirement conventionally calculated, is affected by close control of the heating during relatively short hours scheduled for heating.[5, 6]

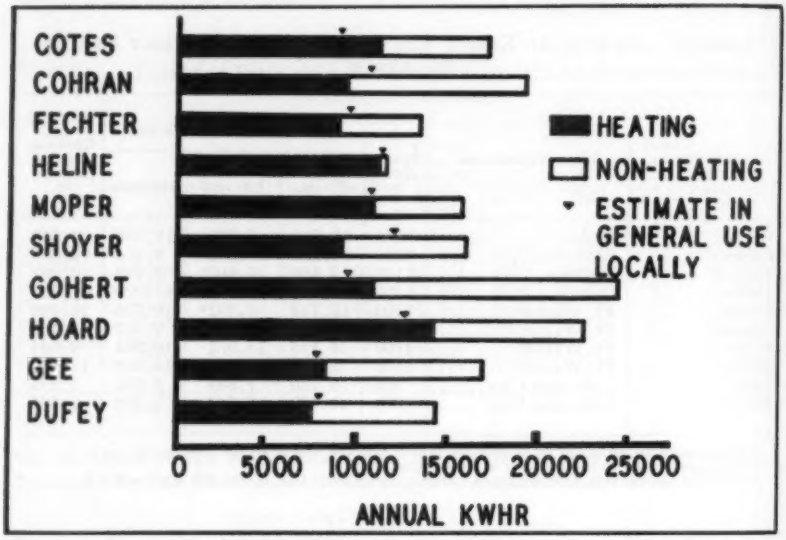

FIG. 3. . . .DIAGRAM OF DATA TAKEN FROM LAST 4 COLUMNS OF TABLE 2 (10 INDIANA HOMES)

The store and offices of a lumber company contain 36,000 cu ft with a calculated heat loss of 148,381 Btuh. Based on 70 deg temperature difference, 37.3 deg average temperature and 6,363 degree days, the calculated energy requirement would be 103,333 kwhr. The estimate on which electric heat was sold assumed 59,752 kwhr to heat. The actual sub-metered energy required to heat it was 19,720 kwhr for the 1955–1956 heating season, which had 6,192 degree days. It is of interest to note that the non-heating use was 109,380 kwhr and the total use was 129,100 kwh.[7]

Acceptance of the following statements as facts logically precedes the conclusions:

1. Efficiency of conversion of electricity to heat with resistance electric heating equipment is 100 percent.

2. Efficiency of utilization of electric heat is high when heaters are located within the space they serve.

3. Internal sources of heat and sun effect contribute materially to satisfying heating requirements in well insulated buildings, regardless of the heating system used.

CONCLUSIONS

1. Use of formulas developed for combustion heating fuel requirements give abnormally high energy requirements for heating with electricity. The reasons for these deviations of actual from calculated values should be investigated.

2. There is need for authentic and realistic evaluation of practical methods for predicting energy requirements for heating with electricity.

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[7] Case history for electric heating (*Electric Heat and Air Conditioning*, May–June 1957, p. 20).

APPENDIX

Table A-2 shows the experience record of average total kwhr billed to all electric home heating customers of the Indiana & Michigan Electric Co. for heating and general use

TABLE A-1. . . .ADDITIONAL DETAILS ON TABLE 2 RESIDENCES

DESCRIPTION	COTES	COHRAN	FECHTER	HELINE	MOPER	SHOYER	GOHERT	HOARD	GEE	DUFEY
TYPE SYSTEM	BB	BB	BB & C	BB	CABLE	BB	CABLE	CABLE	BB	BB
NO. ROOMS	6	7	6	7	6	8	6	—	5	5
INSTALLED KW	9.85	16.9	10.1	8.75	9.8	17.25	11.4	—	8.7	8.7
					TOTAL ENERGY					
Jan.	2788	2368	492	2192	2354	2126	3830	3600	2630	2260
Feb.	2262	2150	606	1690	2090	1380	3456	3250	2290	2220
Mar.	1340	1800	572	780	1322	862	2852	2970	1890	1800
Apr.	618	870	328	178	584	562	2112	3120	1940	1750
May	584	968	362	130	514	552	1348	1300	1050	870
June	516	860	406	56	500	510	1068	970	770	550
July	498	756	900	60	450	574	908	640	570	390
Aug.	432	888	1892	60	382	674	950	670	760	430
Sept.	988	848	2264	470	524	1526	1128	820	610	520
Oct.	1852	1802	2218	1286	1606	2340	1262	1060	810	690
Nov.	2650	2846	2172	2380	2576	2480	2486	1460	1270	1010
Dec.	2482	3082	1644	2458	2816	2136	3396	2600	2140	1910

TABLE A-1 (*continued*). . . . ADDITIONAL DETAILS ON TABLE 2 RESIDENCES

DESCRIPTION	COTES	COHRAN	FECHTER	HELINE	MOPER	SHOYER	GOHERT	HOARD	GEE	DUFFY
TYPE SYSTEM	BB	BB	BB & C	BB	CABLE	BB	CABLE	CABLE	BB	BB
NO. ROOMS	6	7	6	7	6	8	6	—	5	5
INSTALLED KW	9.85	16.9	10.1	8.75	9.8	17.25	11.4	—	8.7	8.7
HEAT ENERGY										
Jan.							2450	2851	1944	1680
Feb.							1976	2505	1628	1550
Mar.							1618	2331	1199	1192
Apr.							958	2411	1198	1158
May							220	624	367	227
June							76	303	186	85
July							4	10	0	0
Aug.							4	7	0	0
Sept.							42	66	54	30
Oct.							208	344	253	177
Nov.							1288	734	581	415
Dec.							2020	1930	1471	1292

TABLE A-2. . . .STATISTICAL EXPERIENCE WITH ELECTRIC HEAT IN INDIANA

MONTH	AVERAGE EXPERIENCE RECORD					
	1954–1955		1955–1956		1956–1957	
	KWHR	HOMES	KWHR	HOMES	KWHR	HOMES
June	1036	1035	786	1053	1054	1806
July	496	1035	587	1101	611	1852
Aug.	412	1035	472	1147	509	1909
Sept.	433	1035	486	1216	538	2033
Oct.	527	1035	594	1268	696	2093
Nov.	1190	1035	1314	1359	932	2213
Dec.	2146	1035	2527	1483	2060	2320
Jan.	2879	1035	3366	1563	2948	2416
Feb.	3170	1035	3211	1622	3434	2492
Mar.	3284	1035	2957	1659	2908	2548
Apr.	2305	1035	2459	1695	2470	2595
May	1246	1035	1860	1746	1781	2663
Total.	19124		20619		19941	

[a] Data based on same residences for full year.

for the periods shown. While no accurate record was kept of the size of the homes, it is estimated the average is about 1150 to 1200 sq ft floor area, which average has been maintained in the region for several years.

DISCUSSION

JOHN EVERETTS, JR., Philadelphia, Pa. (WRITTEN): This paper has aroused considerable argument for and against the use of direct electric heating. The argument is not

primarily against the use of electric heat *per se*; but against the method of calculating the energy requirement as compared to the present standard methods as developed by the Society and as used in heating practice today.

Aside from the figures given in this paper, it was possible to make a study of the Howard C. Shaner residence built in Harvey Cedars, Long Beach Island, N. J., in 1957 and equipped with direct electric heating with each room under its own thermostatic control.

The calculated heat loss by conventional methods is 45,000 Btu per hr. There was a total of 4,485 degree days for the area from October 1, 1957 to April 30, 1958. The fuel oil required, calculated on a conventional basis is 1210 gal. On the basis of 140,000 Btu per gal, this would be equivalent to 50,000 kw of electric energy, neglecting the efficiency of the oil-fired system. At 100 percent efficiency, this would reduce to 35,000 kwhr for 70 percent efficiency for oil.

The occupancy of the house was for 4 months continuous and 8 months from Friday evening to Sunday evening. For the months of October and November 1957, a set-back to 60 F was tried for the unoccupied periods. This was discontinued in early December and all thermostats were left set at 70–72 F with a slight shift to 74–75 F in some rooms during week-end occupancy.

Calculations on the basis of the NEMA Manual for the average of 4,741 degree days showed an annual use of 20,300 kwhr for constant 70 F setting and 12,500 kwhr for 55 F set-back for 5 days per week. The estimated energy cost at 2¢ per kwhr was $406.00 and $250.00 per year respectively. The actual usage with a 60 F set-back for 2 months and no set-back for the balance of the winter for the actual 4,485 degree days at the actual average rate of 2¢ kwhr was $289.00 for an energy use of 14,450 kwhr. This would indicate a very realistic agreement with the NEMA.

It is felt that the problem faced at the moment is not whether electric energy is better or worse than any other fuel, but to take a realistic view of the methods of calculating fuel requirements and it is recommended that this be brought to the attention of the C.O.R. for their action.

FRED CANOVA*, Princeton, N. J.: In the paper the author assumed that the electric heater was located in the room to be heated. How much does the author think an electric heating system centrally located in the house would increase the cost estimates over these given in his paper?

NORMAN J. LIPSTEIN, Schenectady, N. Y.: Since the advantage of electric heating appeared to be the utilization of the heater in the room without any distribution problem, would not a room type of heat pump unit offer additional advantages? The individual room heat pump would provide 2 or 3 times the heat input per unit of electrical energy supplied and still retain the advantages of the room electrical heater.

C. M. ASHLEY, Syracuse, N. Y.: To what extent has the author investigated the effect of sunlight gain through the windows as a possible major explanation of the differences noted between calculations and actual results?

It must be pointed out that as the amount of insulation increases, the proportional effect of the sunlight gain tends to become very much greater and, therefore, it can quite readily account for quite a little of this difference. Also, it is well known that, in general, the experience factor with all kinds of heating on a reasonably well insulated house is considerably less than that which would be predicted by ordinary calculation methods.

AUTHOR'S CLOSURE: On the matter of central heating *versus* room-by-room treatment, using a system where the source of energy is entirely removed from the structure—which is true with electricity and with a steam or hot water system with an entirely isolated boiler room—one of the differences between a central system and the room-by-room

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* Project Engineering, Solar Department, Curtiss Wright Corporation.

treatment is the losses in conveying the heating medium from the central point to the point of use. This can be minimized by suitable insulation of the passage through which the heating medium is carried, whether it be steam, hot water or warm air.

Probably more significant from an economic standpoint is the room-by-room control, which is axiomatic when the heaters are within the room itself, and relatively costly to provide with a central system. But the effect of each room being its own zone will have a significant effect on operating cost. How much depends on the circumstances. It is not meant to imply that it is a matter of 50, 70 or 80 percent, but somewhere in the neighborhood of 10 to 20 percent probably, depending vitally on the amount of sun effect and internal load in the various rooms involved.

With regard to the room type heat pump, it certainly has the same advantages of room-by-room control and each room has its own complete and isolated heating system.

Discussions of the relative merits of heat pumps, and resistance heat with mechanical cooling, and other forms of heating and cooling could go on for hours and probably not reach any acceptable conclusion.

However, in any heating systems or cooling systems, not only operating cost but first cost and amortization of first cost (which is normally by mortgage or bond) is involved. This amortization is just as much an operating cost as the money paid for fuel.

On the question of the effect of sunlight gained through windows it is certainly agreed this would contribute materially to any type of system equally, and it should further be clarified that the author heartily agrees that any insulation applied will have the same effect on the net heat loss of a building regardless of the source of energy for heating. Any heat gained from internal sources and sunlight will be the same regardless of the source of energy for heating.

Actually, none of the measures the electric heat industry has been using are methods that have not been available to all other forms of heating from long ago.

Mr. Everett's comments are quite appropriate and appreciated. They point out that actual results conform much more closely to the NEMA method of calculation than calculation methods recommended for combustion heat. They also point out that actual operating procedures are most likely to deviate from those assumed in early analysis. It is fortunate that the changes in method of operation were noted so that this information has the added value of noting the operational settings actually used.

On Mr. Ashley's latter comments, it has been definitely shown with electric heat that the better insulated the structure, the more substantial the comparative deviation from calculated losses based on the actual construction used. It is well to learn that this same tendency is exhibited with other forms of heating.

Rather than the author alone trying to explain why such deviations occur the Society, and its members, are being asked to give thought to defining an accurate method of calculation for electric energy heating, since it is known from experience that there are such marked deviations from the methods that are now proposed for use.

1664

No. 1664

HEATING BY HYPOCAUST

By EDWIN D. THATCHER*, NEW YORK, N. Y.

WHEREAS the ancient Roman method of heating by hypocaust has for years been correctly recognized as a form of radiant heating, it has also been mistakenly regarded as little more than a quaint archaeological curiosity. Recent investigations have shown it to have been not only a remarkably efficient system of heating for an era lacking the technological aids now taken for granted, but perhaps a more intelligently applied system than present day forms of radiant heating.

THE HYPOCAUST METHOD OF HEATING

The heating medium of the hypocaust was the mixture of smoke, combustion gases and excess air from an open fire, and the hypocaust was essentially an apparatus for drawing off this mixture by natural draft. The apparatus was, in the main, horizontal, in the shape of a hollow floor. At one end of the hollow space was a hearth and at the other was a vertical flue, or flues. In its simplest form the hypocaust provided only a radiant floor. As it developed from its beginnings in the first century before Christ, the number of flues increased. This increment was in part designed to improve the distribution, by natural draft, of the heating mixture within the hollow floor, but when the flues were so multiplied that they covered one or more of the masonry walls of a room, they then provided supplementary radiant surfaces as well. Even the vaulted ceiling was eventually covered, so that a radiant casing was achieved.

The hollow floor was constructed of concrete on a base of 2-ft square, flat tiles supported on pillars of brick about 2 ft-6 in. high and spaced 2 ft on centers in both directions. The flues, when they acted only as draft inducers, were round terra cotta pipes from 8 to 12 in. in diameter. Those that lined the walls and also produced a radiant surface were of rectangular section. From a rather improvisatory beginning, when these were nothing more than roof tiles set with their flanges to the wall, they passed through several stages of development to their

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* Architect, Member of *AIA*.

[1] Exponent numerals refer to References.

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logical culmination as rectangular terra cotta tubes about 15 in. long with average interior dimensions of 3 × 4 in. These tubes were set side by side against the wall, from floor to ceiling, and held in place by mortar and T-shaped iron anchors driven between the tubes and into the masonry behind them. This whole lining was then covered by 4 to 5 in. of mortar, which was often veneered with 1 in. or

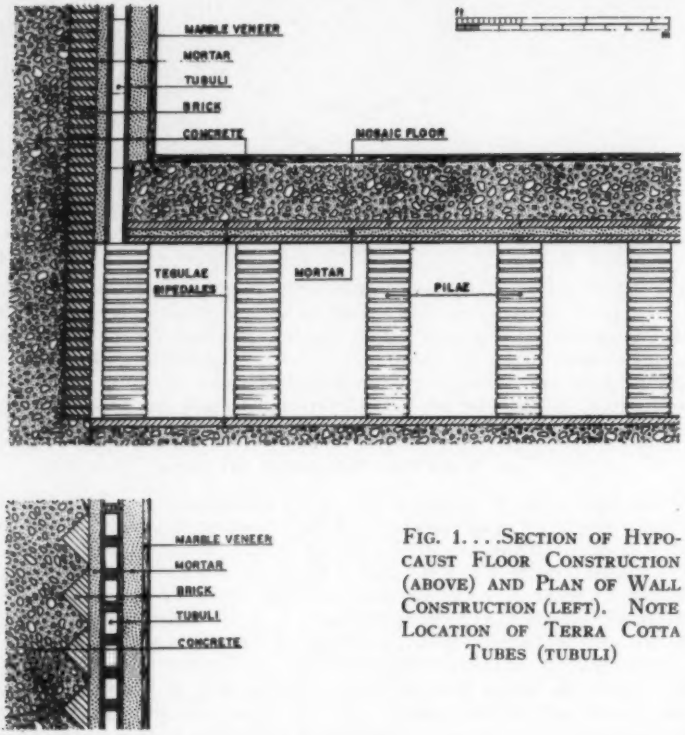

FIG. 1. . . .SECTION OF HYPOCAUST FLOOR CONSTRUCTION (ABOVE) AND PLAN OF WALL CONSTRUCTION (LEFT). NOTE LOCATION OF TERRA COTTA TUBES (TUBULI)

more of marble. Fig. 1 shows a section through a typical hollow floor, as well as a plan of the wall lining.

When the lining of flues was not carried over the vault, it ended at the top of the wall in a horizontal collecting-channel leading to one or more small chimneys that either pierced the roof or ran up obliquely through the wall to its outside face. Exactly how the vault-linings were vented is still open to question because of the scarcity of intact lined vaults among the Roman remains, but there is little doubt that they were vented at or near the apex of the vault, and probably through a system of similar collecting-channels.

Development of wall- and vault-linings did not result in every heated room having a warm wall and ceiling. This development was reserved for those spaces

requiring exceptional warmth, such as the hot rooms of baths, or the living rooms of dwellings in the colder provinces of the Empire. For, although the hypocaust system is most elaborately manifest in the great Imperial Baths, it was widely used for domestic heating in England, Germany and northern France, and even occasionally in Italy. The typical living room usually possessed only a warm floor, with perhaps a lined wall on the weather side. The hot rooms of baths, on the other hand, were jacketed with flues on all their walls and often on their vaults as well.

But in whatever form, the hypocaust provided the ideal means for achieving a condition of comfort in the typical enclosed space of Roman architecture: a high-ceilinged chamber, the volume of which was relatively large in comparison to its floor area. To heat the air volume of such a space would have been highly unpractical, and in the Imperial Baths the task would have been virtually impossible. The hypocaust system, instead, applied itself most directly to the basic condition for human comfort—the inhibition of human heat loss—by providing warm surfaces that inhibited this loss through its 2 broadest avenues: conduction and radiation. By inhibiting the loss in this manner, the system could concern itself only indirectly with the air temperature, or, in extension, with the loss through convection.

It is hardly safe to assume that the Romans had any idea, in the modern sense, of the physiological principles of human heat loss and heat generation, although it would not be too surprising had they had. There is no doubt, however, that they realized the efficacy of using radiant surfaces for comfort, because the development of the hypocaust system, which can be traced over a period of 600 years, shows an increasing awareness of this efficacy.

THE FORUM BATHS AT OSTIA

One of the most dramatic examples of the culmination of this development is to be found in the Forum Baths of the ancient port of Ostia, 15 miles down the Tiber from Rome. As their name implies, these baths lie next to the city's Forum, and were a principal bathing establishment of the community. They were erected in the middle of the second century after Christ and were in use throughout Constantine's time and for possibly 100 years thereafter, or until the barbarian invasions and the end of the Western Empire. They were, therefore, in operation for at least 2 centuries, and perhaps for 3.

The bath building itself is typical of most Roman thermal establishments only in its northern half, where the plan is symmetrically rectilinear. Along the southern facade, on the other hand, are 5 chambers of various design arranged in a stepped pattern that forms a seemingly whimsical contrast. The exterior walls of these 5 chambers, moreover, are almost completely taken up by great windows, and all the archaeological evidence points to their having been unglazed. Since these 5 chambers were the principal heated rooms of the baths, their open windows present a unique problem in both heating and orientation. The success of the solution is implicit in the building's use for over 200 years, but has been confirmed by an inquiry, in modern terms, into the functioning of the building and its heating system[1].

Although the Roman bath essentials are preserved in modern sweating rooms, massage rooms and cold plunges, the special character of the ancient establish-

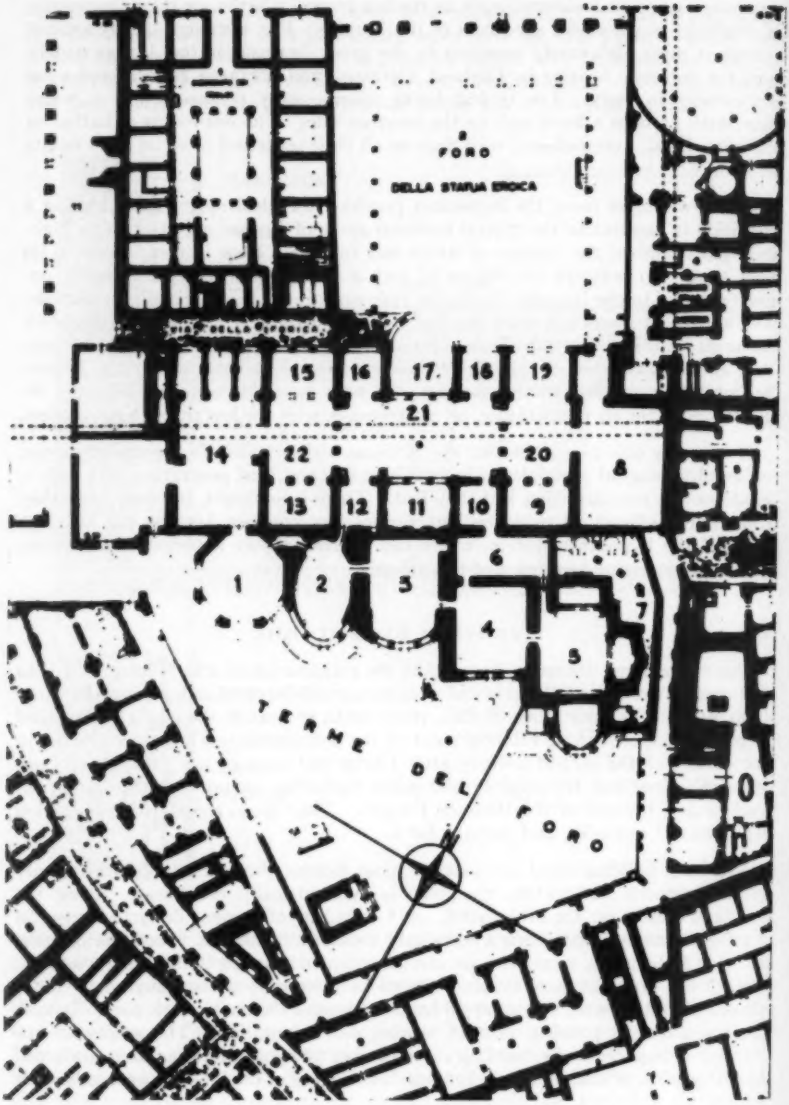

FIG. 2....THE FORUM BATHS (TERME DEL FORO) AT OSTIA AND THEIR SURROUNDINGS

ments has been lost. These were not alone places in which to bathe; they were also the social centers of the community, where a whole day could be pleasantly spent in reading and gossiping, listening to poets and orators, politicizing and, perhaps only incidentally, in exercising and bathing.

A thorough cleansing was the chief end of the bathing; but the therapeutic benefits of steam, warmth and hot water were even then well recognized and were important secondary aims. A good bathing establishment was flexibly arranged so that the bather might follow his own rule. He would have begun by annointing himself. Then he would have started a sweat, either by exercise, by sunbathing, or by lounging about partially clothed in a warm room called a *tepidarium*. He might then have gone into the high, dry heat of the *laconicum*, for a heavier sweat, or have passed directly into the *caldarium*, which was just as hot as the laconicum but, because of its pools of hot water, had a steamier atmosphere. Having sweated enough, he would have scraped off the oil, dirt and sweat with a *strigilis* and might then have soaked for a time in one of the hot pools. After this he could have cooled off slowly by returning to the tepidarium, or he might have taken the hardier course of a cold plunge in the tanks of the *frigidarium*. He could have had a massage before or after the climax in the caldarium, or he might have chosen the laconicum for his climax and finished off with only a cold douche. In any event, he would have favored the establishment that allowed him, with commodiousness, the greatest flexibility; and it is not difficult to understand that, from a purely technical point of view, this place would have had to operate efficiently.

The Forum Baths of Ostia would seem to have been a favored establishment: flexibly planned, commodious and efficient. The general plan of Fig. 2 shows the frigidarium (no. 21) as the center of the northern half of the building and as the focus for nearly all circulation into the hot rooms to the south. It contains 2 cold plunges (11 and 17) and is flanked by 4 *tepidaria* (10, 12, 16 and 18) of which the 2 southernmost are also vestibules into the hot rooms. East and west of this great hall are lounging rooms or libraries (9, 20, 19 and 13, 22, 15) and beyond them the entrance lobbies (8 and 14). The rooms forming the southern facade (1, 2, 3, 4 and 5) are those with great, unglazed windows, previously mentioned. No. 1 is a room for sunbathing—a *heliocaminus*; no. 2 is a laconicum; nos. 3 and 4 are a species of tepidarium; and no. 5 is the caldarium with 3 hot pools, the southernmost thrusting its semicircular window-wall out into the trapezoidal *palaestra*, the exercise yard. This sunny pleasance, a half-acre in area, is bounded on 3 sides by buildings with colonnaded porticoes, that contained shops, snack bars, a latrine, and rooms for lounging or entertainment. Room 6 is another laconicum, and Room 7 is the service area that contained the water heaters for the caldarium pools. From this room a low service passage runs all along the south front of the building, giving access to the hypocaust hearths of Rooms 5, 4, 3, 2 and 1. The vaulted roof of this passage projects several feet above the level of the palaestra and forms a kind of terrace. The more detailed drawings of Fig. 3 show the terrace in plan, as well as the passage in section.

Study of the building's arrangement, with its provisions for various intercommunication, reveals the flexibility of the planning. The skillful disposition of the rooms gave the bather a diversity of choice in his progress to and from the caldarium; and in the rooms themselves, as much as in their disposition, lay the building's luxurious variety as a bathing establishment that undoubtedly attracted a sophisticated and discerning clientele. For the 5 rooms of the south front were

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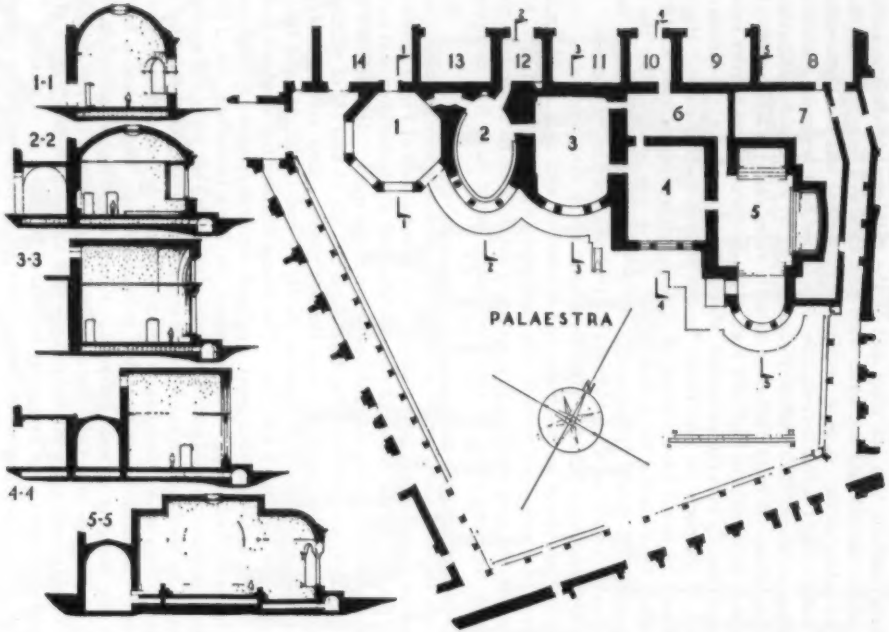

FIG. 3. . . .PLAN AND TYPICAL SECTIONS OF THE 5 ROOMS IN THE SOUTHERN PORTION OF THE BATHS

not only of varied and pleasing form, they were also open to the sun and air, and must have given their occupants a refreshing sense of lucent space and union with the outdoors, while at the same time affording, in their radiant surfaces, secure comfort. The stepped pattern of these rooms is not the result of whimsey; they were successively projected from the facade in order to gain as much sunlight as possible, particularly in the colder seasons of the year. Room 1, as befits a place for sunbathing, shows upon investigation not only the greatest solar gain of all, but also the greatest constancy of sunlight throughout the day. Room 2, with its asymmetrical window, ranks second in this respect. Rooms 3 and 4 receive an almost equal proportion of sunlight, but less than Room 2. Room 5 receives the least of all, and most of this in its semicircular pool. This singularly inviting spot was very likely the most popular place in the Baths. Here the bather was offered the anomaly of outdoor bathing in a steam room; and the combination of cool air, sunlight and hot water, with a steamy refuge close at hand, must have given the ultimate touch of luxury.

ANALYSIS OF THE HEATING SYSTEM

The functioning of the heating system that made these rooms possible is best revealed by an analysis based on that method of radiant heating design which deals chiefly with the regulation of the rate of heat loss from the human body, rather than with the rate of loss from the structure itself[2]. Since this method is primarily concerned with the conditions under which a human being may be comfortable, and only secondarily with the structure that provides these conditions, it is most appropriate for the analysis of an unfamiliar building-type. The analytic procedure consists of the following steps:

Step 1, establishment of probable heating surface temperatures; *Step* 2, derivation of air temperatures from these surface temperatures and from design conditions of wind and temperature; *Step* 3, determination of relative humidities from the derived air temperatures and from design conditions of humidity; *Step* 4, establishment of surface temperatures under various conditions of solar radiation; *Step* 5, calculation of the rate of heat loss of a *nude, normal adult* in the established environment.

If the rate of heat loss of this individual does not exceed his rate of heat generation, the established environment is considered satisfactory. Hence the comfort of the human being is the criterion of the structure's performance in providing a suitable environment. And if one may presume that men have changed very little in the past 1600 years and that the same general requirements for comfort have therefore prevailed, this criterion is the safest to adopt for the analysis of an anomalous ancient structure.

Step 1: In order to proceed with *Step* 1 and determine the probable heating surface temperatures, it was decided to set the values of the thermal conductivities, k, of the various materials in the structure—such as marble, brick and tile—at the same values used for these materials today. For the dense concrete of the floor, containing limestone aggregate, a k of 11 Btu was assigned. For the wall and vault concrete, containing tufa aggregate, the k-value was taken as 7. The surface conductivity, f, of the interior building surfaces was set at 1.65, for still air; that for the exterior building surfaces was taken as 6.0 for a wind velocity of 15 mph.

The analysis then proceeded by assuming an average floor temperature of 100 F. The fires in the hypocausts would have formed a more obvious basis, but the scant information concerning them is quite inadequate. On the other hand, there is a surprising amount of evidence in Latin literature regarding the heat of the baths, the appraisal of which makes the assumption of 100 F a conservative one. This is further confirmed by the temperatures of modern Turkish baths, as well as by the fact that 100 F is not at all intolerable to bare feet.

With this floor temperature, and using the Equations A-1 to A-5 as stated in the Appendix A, the temperatures of the wall and vault surfaces in each of the 5 rooms were calculated. They were found to be in the order of 102 to 110 F for the walls and 93 to 100 F for the vaults.

One of the operations (use of Equation A-1) in this part of the procedure of *Step* 1 indicated that the temperatures of the gaseous mixture of smoke, combustion gases and excess air under the floor was of the order of 125 to 300 F, while the corresponding temperatures in the tubes of the walls and vaults was of the order of 105 to 230 F.

Step 2: *Fixing the design air temperatures.* The first operation was to arrive at the design conditions of wind and outside air temperature, and this was done by studying the meteorological data for Rome at the present time. This is believed suitable for there is no evidence to show that the Roman climate of today is essentially changed from that of ancient times. As a result of this study (based on 1953) the following design conditions were arrived at: (1) a temperature of 30.2 F, with wind at 10 mph from the North; (2) a temperature of 37 F with wind at 7.1 mph from the southwest.

The *inside air temperature* in each of the 5 rooms was then regarded as that temperature attained by an initial mixture of inside air and infiltrating air upon exposure to the warm surfaces of the room. The temperature was calculated by means of Equation B-1 of Appendix B, with its corollary Equations B-2, B-3 and B-4, and which permitted a simple calculation of the infiltration of outside air.

This method of calculating the infiltration is based on the belief that air movement within the rooms would have taken 2 principal directions: (1) in a more or less horizontal path between openings at the same general level; (2) in an ascending path from a low inlet to a high outlet.

Aside from the windows, the openings in the rooms consisted of doorways and of either high windows in the lunettes of the north walls or circular openings, called *oculi*, in the crowns of the vaults. The free area of these latter was regulated by means of a metal disc, rigged on chains directly beneath, so that it could be raised for complete closure or lowered for the maximum opening. The doorways were equipped with doors on pivot hinges and the high windows with hinged shutters.

With a southerly wind, there would have been air movement, due to wind pressure alone, from the windows to the doors at the north end of the room. There would also have been an exfiltration through the *oculus* or the high north window, due to a combination of wind pressure, the difference in temperature between outside and inside air, and the height from inlet to outlet. With a northerly wind, there would have been a generally upward air movement, from the doors at the north end to both the *oculus* and the upper part of the south windows. This movement would have been governed by a combination of wind pressure, temperature difference and height. Air coming through the high north windows under

such conditions would have dropped until warmed by the air of the room, but its movement, governed principally by wind pressure, would still have been southerly.

Each of the rooms was examined under 5 conditions of infiltration, *viz*:

1. with a northerly wind—doors and oculus closed;
2. with a northerly wind—one door open and oculus closed;
3. with a southerly wind—doors and oculus closed;
4. with a southerly wind—one door open and oculus closed;
5. with a southerly wind—doors closed and oculus opened.

The velocity of air motion within the rooms was established only for its effect upon the *nude normal adult*. The additional heat gained from the room surfaces by moving air was disregarded. Air motion greater than 100 fpm was regarded as causing discomfort; that less than 50 fpm was regarded as ideal. The air speed within the room, V_m, was determined by the equation

$$V_{ot}/V_m = Q_m/Q_{ot} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

V_{ot} = velocity of infiltrating air, feet per minute.
Q_{ot} = volume of infiltrating air, cubic feet per minute.
Q_m = volume of air in the room, cubic feet per minute = $Q_{ot} + Q_i$, in which
Q_i = volume of inside air, cubic feet per minute.

Step 3: *Determination of Relative Humidities*. The relative humidity of the final, heated mixture of inside and infiltrated air—the air at the attained temperature (t, in the Equation B-1)—was established by the equation

$$\phi = p_w/p_s \quad . \quad (2)$$

where

ϕ = relative humidity, percent.
p_w = partial pressure of water vapor, inches of mercury.
p_s = saturation pressure of water vapor, inches of mercury.

The design conditions for relative humidity were established from an average of the recorded relative humidities for January and February for Rome. A study of the meteorological data shows these months as the most humid of the 5 winter months; as might be expected, it also shows the highest relative humidities when southerly winds blow from the Mediterranean. A relative humidity of 63 percent was established as a design condition with north winds; 78 percent was established as a design condition with southwest winds.

Step 4: *Establishment of Surface Temperatures under Various Conditions of Solar Radiation*. The effect of solar radiation on the room surfaces was expressed as an increase in the radiant temperature of the room as a whole. This increase was obtained by regarding both the reflected and emitted solar radiation as additional heat emitted from the surface and by then computing the corresponding surface temperature for the increased emission. This temperature then represented the increased radiant temperature. The heat emission under these conditions—the black body radiation—was found by use of Equation C-1 of Appendix C.

Solar radiation, for horizontal and variously oriented vertical surfaces, was computed by use of Equation C-2 of Appendix C.

Values for direct solar radiation only were used in the computations because of the uncertainty of accurately calculating the effect of diffuse solar radiation upon the interior of a building. These values, moreover, were deduced for only cloudless (clear sky) conditions and for average conditions, a postulate that simplified the calculations without materially affecting the results.

Cloudy conditions were considered to allow of no direct solar radiation at all. Hourly values were considered necessary because of the evident limitation of use of the bathing rooms, for the references in ancient literature point to a customary bathing period from noon to 17:00, with its peak at about 14:30. Values were, therefore, deduced for the hours of 10:00, 12:00 and 14:30, under both clear sky and average conditions, and for December 21, March 21, June 21 and September 21. Shadows were cast upon a plan of the rooms at these 4 significant dates of the year and the areas of sunlight thus derived were used in the calculation of total radiation values. In the analysis, however, only the values for December 21 were used as a design condition because the unit radiation on this date is the lowest of the year. The total effective direct solar radiation, in respect to the room surfaces, was taken as the sum of reflected radiation and of previously absorbed radiation that was being emitted from surfaces of an absorptivity of 0.5 after a time lag of approximately 2 hours, a period that seemed consistent with the composition of the floors and walls and with their internal temperatures. Thus, for example, the total effective radiation at 12:00 was the sum of the reflected radiation at that hour and the absorbed radiation at 10:00. The effect of solar radiation was calculated for the interior of the rooms, only; no attempt was made to estimate the heat gain through walls and vaults by the action of the sun on the exterior of the building.

The results of the first 4 steps in the analysis thus served to fix the environmental conditions surrounding the bather, and consisted of surface temperatures, air temperatures, air speeds and relative humidities. They made it possible to proceed to *Step* 5.

Step 5: *Calculation of the Heat Loss Rate of a Nude Normal Adult in the Bath Environment.* The loss through radiation to the surroundings was calculated by the use of Equation D-1, while the losses through convection in both still and moving air at 50 percent relative humidity, were determined from Equations D-2 and D-3 of Appendix D.

The human loss through evaporation was approximated by a synthesis of experimental results gathered from nude, lightly clothed and normally clothed subjects in still air of 50 percent relative humidity. At humidities lower than 50 percent additional losses were approximated by the use of effective temperatures and a psychometric chart, as well as by Equation D-2 for convective loss in still air.

Human loss through conduction was disregarded on the assumption that the bather was standing at all times and was wearing bath shoes, or clogs.

The human subject chosen for the analysis—*nude normal adult*—was considered to be 5 ft - 8 in. tall, to weigh 155 lb and to have a body surface area of 19.5 sq ft. And although he was assumed to be standing, his metabolic rate of heat generation was taken as 400 Btu per hour, instead of the more usual 430 Btu. The lower rate was chosen as a factor of safety.

PRESENTATION OF RESULTS

Results of the analysis of the system were worked up into tables—one for each of the 5 rooms—and one of which is included here as Table 1. In it, the reactions of the room under the calculated design conditions are in terms of human heat loss, but in a manner that emphasizes what might loosely be called heat *gains, i.e.*,

TABLE 1—TABULATION OF THE CALCULATED PERFORMANCE OF ROOM 3 WITH HEATED VAULT. A LIKE TABLE WAS PREPARED FOR EACH OF THE 5 ROOMS

Conditions of Infiltration	Time of Day	Conditions of Air			Cloudy Sky		Solar Radiation[a]						Notes
							Average			Clear Sky			
		Temp, F	Speed, fpm	Rel. Humid., %	Mrt, F	Human Loss, Btu	Mrt, F	Human Loss, Btu		Mrt, F	Human Loss, Btu		
								1	2		1	2	
1	10:00 12:00 14:30	82	18	9	88	**353**							
2	10:00 12:00 14:30	63	98	18	82	640	85 86 84		**316** **269** **343**				(*a*)
3	10:00 12:00 14:30	80	14	17	88	**353**							
4	10:00 12:00 14:30	53	124	43	80	800				86 89 85		**162** **60** **228**	(*b*)
5	10:00 12:00 14:30	65	48	28	83	585	86 87 85		**263** **216** **290**				

[a] The loss in Column 1 is for the bather standing out of the sun, and in Column 2 for the bather in the sun.

Note (*a*): Air motion near the maximum; sweating; possible discomfort. *Note* (*b*): High air motion plus probably heavy sweating; discomfort. Average of bold face figures is plotted in Fig. 4.

losses below the metabolic rate of 400 Btu per hour. These losses below the metabolic rate, however, do not here signify excessive storage of heat in the body. For in cases of apparently small loss, this loss would have been increased by heavy sweating, a condition not wholly reflected in the method of calculating evaporative losses. And in actual practice, additional ventilation or the banking of the fires would have been employed to raise the total human loss to a supportable rate. The computed values of human loss result from the nature of the examination, which was based for all rooms on one set of premises and one set of design conditions. These values were therefore chosen as a means of expressing the reactions of the rooms, not because they represent an altogether accurate physiological picture of the plight of the *nude normal adult*, but because they offer a common,

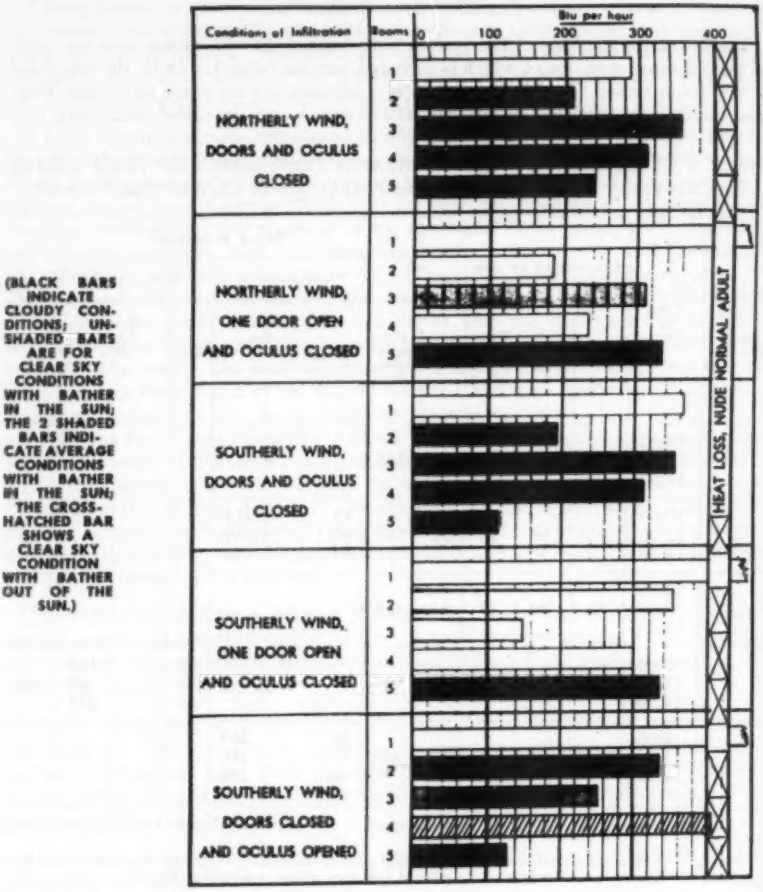
Conditions of Infiltration
Rooms
Btu per hour
0
100
200
300
400
NORTHERLY WIND, DOORS AND OCULUS CLOSED
NORTHERLY WIND, ONE DOOR OPEN AND OCULUS CLOSED
SOUTHERLY WIND, DOORS AND OCULUS CLOSED
SOUTHERLY WIND, ONE DOOR OPEN AND OCULUS CLOSED
SOUTHERLY WIND, DOORS CLOSED AND OCULUS OPENED
1
2
3
4
5
HEAT LOSS, NUDE NORMAL ADULT
(BLACK BARS INDICATE CLOUDY CONDITIONS; UNSHADED BARS ARE FOR CLEAR SKY CONDITIONS WITH BATHER IN THE SUN; THE 2 SHADED BARS INDICATE AVERAGE CONDITIONS WITH BATHER IN THE SUN; THE CROSS-HATCHED BAR SHOWS A CLEAR SKY CONDITION WITH BATHER OUT OF THE SUN.)

FIG. 4. . . .HEATING PERFORMANCE OF ROOMS UNDER VARIOUS EXTREME AND AVERAGE CONDITIONS IN TERMS OF HUMAN HEAT LOSS

human denominator for describing and comparing rooms that have all been examined under the same criteria. Fig. 4 is a synthesis of the results from the tabulation for each room.

COMMENTS AND SOME IMPLICATIONS OF THE STUDY

Although the discovery of many technical details of hypocaust operation must wait upon further research, the analysis here described did disclose enough of the

system's functioning to demonstrate its ability to achieve a condition of comfort in spaces of large volume open to the outer air, and confirmed the success of the solution of a problem that would be considered unique even by modern standards. The success of the solution rested largely upon the heating system, but it also required the support given by skillful planning in the manipulation of the local climate.

The buildings along the southeast and southwest sides of the palaestra were of only 2 stories—high enough to act as windbreaks but low enough to admit the winter sun. The building on the northeast was at least 3 stories high and gave the most protection against the worst weather. The principal entrance to the palaestra—from the Forum—was in the westerly corner. This broad opening allowed the afternoon sun to reach Room 1, but it also provided the only real ingress for occasionally bad southwesterly weather. Since such weather, in winter, is rare for the locality and generally cloudy, it would not have permitted sunbathing, and would therefore not have found the heliocaminus in use; at such times this room would have acted as a windbreak for its neighbors to the east. This secondary function of the heliocaminus serves to emphasize it as a room more characteristic of the skillful planning that typifies the whole complex than any other single space therein. Alone of the 5 rooms, it depended upon sunlight for successful functioning, for it had only a radiant floor and was not protected on the north by spaces of intermediate temperature, such as Rooms 10 and 12. On clear days it enjoyed almost constant sunlight and, properly enough, received the greatest amount of solar radiation during the winter months. The clear winter days, however, were almost invariably those of northerly weather, when the infiltration from the northwest side of the room was offset by a maximum of solar radiation. A protective vestibule on this side was therefore unnecessary. The orientation and fenestration of the room were very nearly ideal, and this is finally confirmed by the fact that the room was usable for sunbathing by *nude normal adults* on 67 percent of the days from November to March, when only 68 percent of the days in this season had any sunlight at all. In other words, the room was usable 98 percent of the time.

Whereas the other 4 rooms along the southern facade were designed to utilize solar radiation, they all could function without it. Their planning, however, did not neglect the need for protection against the north. Rooms 10 and 12, with radiant floors and side walls that received surplus heat from the hypocausts of Rooms 6 and 3, respectively, acted as vestibules to Rooms 6 and 2. And Room 6, in turn, was an additional buffer for Rooms 4 and 5. The semi-circular vestibule between Rooms 1 and 2 contained a radiant floor and north wall and effectively screened the laconicum from the cool air of the heliocaminus.

Since the primary purpose of the analysis was to determine whether or not the 5 open rooms of the Forum Baths could, with unglazed windows, have provided a comfortable environment, it did not reveal all of the technical details of hypocause operation. Some insight into the nature of the fires that produced the heating medium of smoke, combustion gases and excess air was obtained, however, in the calculations of this mixture's velocity in the hypocaust and in the tubes. The relatively low gas temperatures—125 to 300 F in the hypocaust and 105 to 230 F in the tubes—implies not only a small mass flow of gas but also a relatively small fire. It has generally been assumed that the hypocaust was an inefficient apparatus in that it required quantities of fuel out of proportion to the heat it produced. This evidence of a small fire, however, gives point to the several implications in

ancient literature of a *gentle fire*, and agrees with the findings of Fritz Kretzschmer, the German heating engineer who operated a reconstructed hypocaust in the Saalburg[4]. The chances are, therefore, that the hypocaust was a comparatively economical apparatus, which should be no great surprise to those who have computed fuel costs for properly designed modern radiant heating systems.

The implications of this study and analysis for modern heating practice need not be historical only. That the Forum Baths provide an ancient example of the advantages of proper orientation and of a respect for the conditions of the climate does not make that example less pertinent to modern practice, where these advantages are in danger of neglect through a mistaken reliance on gadgetry. Nor does the age of these ruins lessen the strength of their demonstration that there is more perception, logic and efficiency in using natural forces than in unnecessarily resisting them. Most important of all, the Forum Baths, in an almost dramatic fashion, testify to the efficacy of radiant heating, particularly in spaces of large volume. And they emphasize the signal quality of this heating method, which is its direct application to the essential problem of making people comfortable. Faced with the complex requirements of modern architecture, and often with a lack of cooperation from architects themselves, it is understandable that heating design should be almost forced into time-saving techniques that emphasize building heat loss at the expense of more pertinent human factors. But with current technological advantages, modern practice should be able to do better—to do at least as well as the Romans who, by accident or not, made man the measure of their purposes.

ACKNOWLEDGMENTS

The author acknowledges the courtesy of the American Academy in Rome in granting its permission to make use of the illustrations here presented.

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APPENDIX A

MATHEMATICAL BASIS FOR STEP 1.—ESTABLISHMENT OF PROBABLE HEATING SURFACE TEMPERATURES

As stated, an average floor temperature of 100 F was assumed, based on consideration of tolerable floor temperatures and on evidence found in Latin literature.

With the assumed floor surface temperature of 100 F it was possible to approximate closely the temperature of the mixture of smoke, combustion gases and excess air under the floor by means of the equation

$$\frac{R_{so}}{R_s} = \frac{t_a - t_{so}}{t_a - t_h} \quad \text{(A-1)}$$

where

R_{so} = resistance of the floor surface.
R_s = total resistance of the floor structure.
t_a = air temperature of the room, Fahrenheit degrees.
t_{so} = temperature of the floor surface = 100 F (by assumption).
t_h = temperature of the gaseous mixture under the floor, Fahrenheit degrees; *i.e.*, the hypocaust temperature.

From the temperature of this mixture, the surface temperatures of the walls or vaults were found by means of the similar equation

$$\frac{R_{wo}}{R_{wi}} = \frac{t_a - t_{wo}}{t_a - t_t} \quad \text{. . . . . (A-2)}$$

where

R_{wo} = resistance of the wall or vault surface.
R_{wi} = total resistance of the material separating the interior of the terra cotta tubes from the room.
t_a = air temperature, as in Equation A-1.
t_t = mean temperature of the gaseous mixture in the tubes, Fahrenheit degrees.
t_{wo} = temperature of the wall or vault surfaces, Fahrenheit degrees.

In both Equations A-1 and A-2, the resistances of the surfaces—,R_{so} or R_{wo},— are equal to

$$\frac{t_a - t_{so}}{q_{ro} - q_{co}} \text{ or } \frac{t_a - t_{wo}}{q_{ro} - q_{co}} \quad \text{. . . . . (A-3)}$$

where

q_{ro} = heat output of the surface through radiation, Btu per (hour) (square foot) = $\sigma F_a F_e (T_{so}^4 - T_m^4)$:
σ = 1730 × 10^{-12} Btu per (hr) (sq ft) (Rankine deg)^4 = Stefan-Boltzmann radiation constant.
F_a = geometrical factor accounting for the shape and relative position of the room surfaces = 1.
F_e = emissivity factor accounting for the characteristics of the room surfaces for absorption and emission of radiation = $1/[(1/e_1) + (1/e_2) - 1]$ = 0.82; e_1 and e_2 were both taken as 0.9.
T_{so} = temperature of the surface, Rankine degrees.
T_m = mean temperature of all other room surfaces, Rankine degrees.
q_{co} = heat output of the surface through convection, Btu per (hour) (square foot) = $h_c(t_{so} - t_a)^{1.25}$:
h_c = unit convective conductance, Btu per (hour) (square foot) (Fahrenheit degree temperature difference) = 0.48 for floor surfaces, 0.354 for wall surfaces, and 0.3 for vault surfaces.

Also in both Equations A-1 and A-2, the total resistances of the structure,—R_s or R_{wi},—are the reciprocals of the coefficients of heat transmission,—U_s or U_{wi}—, which are equal to $1/[(1/f_{so}) + (x/k) + (1/f_{si})]$

where

f_{so} = surface conductance of the surfaces = $1/R_{so}$ *or* $1/R_{wo}$.
x = thickness, in inches, of the structure.

k = conductivity of the structure.
f_{si} = surface conductance of the interior surfaces of the hypocaust or the tubes.

In the case of the hypocaust, $f_{si} = h_r + h_c$, in which

h_r = unit thermal radiative conductance, Btu per (hour) (square foot) (Fahrenheit degree), and was computed from tabular values of h_r/F_aF_e, with $F_a = 1$ and $F_e = 0.82$.
h_c = average unit thermal convective conductance at the position, x, from the leading edge of the surface, Btu per (hour) (square foot) (Fahrenheit degree) $= 2h_{cx}$:
$h_{cx} = 0.0562\,[(T_h) + (T_{si})/2]^{0.3}(vd)^{0.8}/x^{0.2}$, in which
v = velocity of the gaseous mixture, feet per second.
d = density of the gaseous mixture, pounds per cubic foot.
x = dimension, in feet, from the leading edge of the surface to the position, x, which was taken as that end of the hypocaust opposite the hearth.
T_h = hypocaust temperature, Rankine degrees.
T_{si} = surface temperature of the interior surface of the floor structure, Rankine degrees $= t_{si} + 460$; t_{si} may be found by the equation, $R_{cx}/R_a = (t_a - t_{si})/(t_a - t_h)$, in which $R_{cx} = 1/U_{cx}$, and $U_{cx} = 1/[(1/f_{so}) + (x/k)]$.

In the case of the tubes, $f_{si} = (0.32v)^{0.8}/D^{0.25}$, in which

v = velocity of the gaseous mixture, feet per second.
D = interior equivalent diameter of a tube, feet.

In Equation A-2, for the surface temperatures of walls or vaults, the mean temperature of the gaseous mixture in the tubes, t_t, is equal to

$$\frac{t_1 + t_2}{2} \quad \ldots\ldots\ldots\ldots \quad \text{(A-4)}$$

where

t_1 = temperature of the mixture entering the tube, Fahrenheit degrees, which is the same as the hypocaust temperature, t_h.
t_2 = temperature of the mixture leaving the tube $(t_1 - t_b)/e^z + t_b$:
t_b = temperature of the air surrounding the tube, Fahrenheit degrees $= 2[(t_1 + t_2)/2) + 3t_a]/5$.
e = 2.718 = Naperian Base for logarithms.
$z = ((U_{w1}(3P/10) + U_{w2}(3P/10) + U_{w3}(4P/10)L)/14.4\,AdV$:
$U_{w1} = 1/R_{w1}$.
U_{w2} = overall coefficient of heat transmission for the structure behind the tube $= 1/[(1/f_{si}) + (x_{w2}/k_{w2}) + (1/f_o)]$, in which
f_{si} = surface conductance, as above.
x_{w2} = thickness, in inches, of the structure behind the tube.
k_{w2} = conductivity of this structure.
f_o = surface conductance of the opposite side of this structure.
U_{w3} = overall coefficient of heat transmission for the material separating one tube from another $= 1/[(1/f_{si}) + (x_{w2}/k_{w2}) + (1/f_{si})]$, in which
x_{w3} = thickness, in inches, of the material.
k_{w3} = conductivity of this material.

P = interior perimeter of the tube, feet.
L = length, or height, of the tube, feet.
A = cross-sectional area of the tube interior, square feet.
d = density of the mixture at temperature, t_t, pounds per cubic foot.
V = mean velocity of the mixture at temperature, t_t, feet per minute.

The velocity of the gaseous mixture, in both the hypocaust and in the tubes, was estimated by use of the equation

$$v = \frac{w}{Ad} \quad \ldots \ldots \ldots \ldots \quad \text{(A-5)}$$

where

v = velocity of the mixture, feet per second.
w = mass flow of the mixture, pounds per second, estimated from tabular values of flow rates in modern chimneys under temperature conditions similar to those found, by calculation, in the hypocaust and in the tubes.
A = cross-sectional area of the flow, square feet, *i.e.*, the free cross-sectional area of the hypocaust or of a tube.
d = density of the mixture, pounds per cubic foot, which was taken as equal to the density of dry air at standard barometric pressure (29.921 in. of Hg).

APPENDIX B

MATHEMATICAL BASIS FOR CALCULATING INSIDE AIR TEMPERATURE

In *Step 2* of the analytic procedure, the inside air temperature was regarded as that temperature attained by an initial mixture of inside air and infiltrating air upon exposure to the warm surfaces of the room. It was determined by the equation

$$(t - t_m) = \frac{H}{0.24 Q_m d} \quad \ldots \ldots \ldots \quad \text{(B-1)}$$

where

t = attained inside air temperature, Fahrenheit degrees.
t_m = temperature of the initial mixture of inside and infiltrating air, Fahrenheit degrees.
H = total heat output from room surfaces at temperature, t_m, Btu per hour.
Q_m = volume of the mixture of inside and infiltrating air, cubic feet per hour.
d = density of the initial mixture of air at temperature, t_m, pounds per cubic foot.
0.24 = specific heat of air, Btu per (pound) (Fahrenheit degree).

$$t_m = \frac{Q_{oi} T_o + Q_i T_i}{Q_m} - 460^\circ \quad \ldots \ldots \ldots \quad \text{(B-2)}$$

where

Q_i = volume of inside air, cubic feet per hour.
T_i = temperature of inside air, Rankine degrees.
T_o = temperature of infiltrating air, Rankine degrees.
Q_{oi} = volume of infiltrating air, cubic feet per hour = $Q_{oh} + Q_{on}$; in which
Q_{oh} = volume of infiltrating air, cubic feet per hour, due to the temperature difference between the outside and inside air and the height from inlets to outlets = $Q_{oh1} + R_2$:
$Q_{oh1} = 60CA\sqrt{h(t_i - t_o)R_1}$, in which

C = constant of proportionality and effectiveness of openings = 9.4 for favorable conditions and 7.2 for unfavorable conditions.

A = free area of inlets or outlets, whichever is the smaller, square feet.

h = height from inlets to outlets, feet.

t_i = temperature of inside air, Fahrenheit degrees.

t_o = temperature of outside air, Fahrenheit degrees.

R_1 = a factor determined by the ratio of inlet area to outlet area, or vice versa, and accounting for the increased infiltration caused by the excess of one area over another = 1.5.

R_2 = a factor determined by the ratio of that infiltration caused by temperature difference to the total infiltration calculated for both temperature difference and wind pressure; this ratio may be expressed as $Q_{oh1}/(Q_{oh1} + Q_{oh2})$, in which

Q_{oh2} = volume of air, cubic feet per hour, forced through the same openings by the wind = $60\ EA\ V$:

E = a factor for the effectiveness of openings, with values of 0.5 to 0.6 for perpendicular winds, and of 0.25 to 0.35 for diagonal winds.

A = area, as above.

V = wind velocity, feet per minute.

Q_{oa} = volume of infiltrating air, cubic feet per hour, forced by the wind through openings at the same relative level = $60\ EA\ VR_1$.

$$H = q_{cs}A_s \quad \text{. . . . . (B-3)}$$

where

q_{cs} = heat output by convection from the floor, wall and vault surfaces, as found above in Step 1.

A_s = area of these surfaces, square feet.

$$Q_m = Q_{ot} + Q_i \quad \text{. . . . . (B-4)}$$

where

Q_i = volume of inside air, cubic feet per hour.

Q_{ot} = volume of infiltrating air, cubic feet per hour.

APPENDIX C

ESTABLISHMENT OF SURFACE TEMPERATURES UNDER VARIOUS CONDITIONS OF SOLAR RADIATION, STEP 4

As stated, the Equation C-1 which follows was used to establish these temperatures:

$$q = e\sigma T^4 \quad \text{. . . . . (C-1)}$$

where

q = heat emission, Btu per (hour) (square foot).

e = emissivity of the surface = 0.9.

σ = 1730×10^{-12} Btu per (hr) (sq ft) (Rankine deg)^4 = Stefan-Boltzmann radiation constant.

T = temperature of the surface, Rankine degrees.

Equation C-2 was used for computing solar radiation for horizontal and variously oriented vertical surfaces.

$$I_D = KI_{Dn} \quad \text{(C-2)}$$

where

I_D = the direct solar radiation, Btu per (hour) (square foot of surface).
I_{Dn} = the direct solar radiation normal to the sun's rays, Btu per (hour) (square foot of surface).
$K = \cos\theta$ = cosine of the angle of incidence:
$\cos\theta = \sin\beta$ = the sine of the solar altitude, for horizontal surfaces.
$\cos\theta = \cos\beta\cos\gamma$, for vertical surfaces:
γ = the wall solar azimuth = $\Phi \pm \psi$ ($\pm$ depending upon the orientation of the wall and the time of day):
Φ = the solar azimuth.
ψ = the wall azimuth.

Clear sky conditions were assumed to prevail when the cloud-cover was not greater than 2; average conditions when the cloud-cover was between 3 and 7; cloudy conditions when it was between 8 and 10. The values for clear sky conditions were obtained by equating daily values for the region with hourly values for the latitude; those for average conditions were obtained by equating the hourly heliophany with a daily radiation value and thereby deriving an hourly radiation value*.

APPENDIX D

BASIS FOR DETERMINATION OF HUMAN HEAT LOSSES—STEP 5

In *Step 5* of the analytic procedure, the human loss through radiation was determined by use of the equation

$$q_{rh} = \sigma AE(T_1^4 - T_2^4) \quad \text{(D-1)}$$

where

q_{rh} = heat loss, Btu per hour.
$\sigma = 1730 \times 10^{-12}$ Btu per (hr) (sq ft) (Rankine deg)^4 = Stefan-Boltzmann radiation constant.
A = body surface area = 19.5 sq ft.
E = a factor accounting for both the shapes and the relative positions of the body and of the room surfaces surrounding it, as well as for the characteristics of both body and room surfaces for absorption and emission of radiation =
$1/[(1/e + (A/A_r)(1/e_r - 1)] = 0.9$, in which
A = body surface area, square feet.
A_r = room surface area, square feet.
e = emissivity of the body surface = 0.9.
e_r = emissivity of the room surfaces = 0.85.
T_1 = temperature of the body surface = 553 Rankine deg.
T_2 = mean radiant temperature of the room surfaces, Rankine degrees.

* E. D. Thatcher: The open rooms of the Terme del Foro at Ostia (*Memoirs of the American Academy in Rome*, Vol. 24, 1956, p. 236); see also *Application of Climatic Data to House Design* (U. S. Housing and Home Finance Agency, January 1954, p. 93); HEATING VENTILATING AIR CONDITIONING GUIDE (AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Vol. 31, 1953, p. 272); Gaetano Vinaccia: Il corso del sole in urbanistica ed edilizia, 1939, p. 129; Gaetano Vinaccia: L'orientazione stradale equisolare (*Case d'oggi*, Vol. 4, 1942, p. 1); Pericle Gamba: Il sole a Roma (*Atti del IV° Congresso Nasionale di Studi Romani*, Vol. 5, 1938, p. 52).

The human losses through convection, in both still and moving air, at 50 percent relative humidity, were determined by use of the equations

$$q_{ch} = CA\left(\frac{1}{d_b}\right)^{0.2}\left(\frac{1}{T_f}\right)^{0.181}(t_b - t)^{1.27}, \quad \text{for still air} \quad \text{(D-2)}$$

$$q_{ch} = h_c A(t_b - t), \text{ for moving air} \quad \text{(D-3)}$$

where

q_{ch} = heat loss, Btu per hour.
C = a constant for the shape of the surface; *i.e.*, a vertical cylinder = 1.235.
A = body surface area = 19.5 sq ft.
d_b = diameter of a cylinder 5 ft 8 in. high with a surface area of 19.5 sq ft = 13.15 in.
T_f = average of body surface temperature and surrounding air temperature, Rankine degrees.
t_b = temperature of the body surface = 93 F.
t = temperature of the surrounding air, Fahrenheit.
h_c = unit convective conductance, Btu per (hour) (square foot) (Fahrenheit degree temperature difference) = $0.211\ (T_f)^{0.43}\ (vd)^{0.8}/D^{0.4}$ in which
v = air velocity, feet per second.
d = density of the air at temperature, t, pounds per cubic foot.
D = cylinder diameter, feet.

DISCUSSION

F. J. LINSENMEYER, Detroit, Mich.: The author described the construction of these buildings and he may have some information on the minimum temperatures in Ostia. Can he give more information as to what winter temperatures were encountered and what possible inside temperatures might have been in the baths or in Roman homes that were heated with this radiant heat? In other words, can present ideas of comfortable inside temperature be compared with those of the Romans?

W. O. HUEBNER, New York, N. Y.: I enjoyed the author's paper, and listened with great pleasure to his presentation, and am happy that the Society has seen fit to put a paper of this type on the program.

All are inclined to see the progress of our art mainly from our own point of view, and an historical study may sometimes serve to place present-day opinions in a different light. Therefore, I agree with Mr. Thatcher when he states that the implications of the study need not be historical only. I, too, am convinced that it is frequently possible to provide a comfortable environment in a building without *mistaken reliance on gadgetry.*

However, I cannot follow the author in believing that heating design today is *forced into time-saving techniques that emphasize building heat loss at the expense of more pertinent human factors.* I think air-conditioning systems of large buildings as well as heating systems for the average development house today provide a much more comfortable and healthful indoor environment than the average shelter provided during the last 2000 years of human history. Also, we hardly need fear comparison with the Romans, whose society was based on a slave economy. They were concerned with comfort for the few only; we are concerned with comfort for the many.

AUTHOR'S CLOSURE: Because of what Mr. Heubner said regarding comparison with the Romans, I think that perhaps I did not make myself clear. There is no thought of advocating that we imitate the Romans in any technical sense, but only in their application of this method of heating. I got a great deal of inspiration, if you will permit the expression, from the way in which they applied this method—applied it in spite of the technological disadvantages of their time.

A building of this nature, which is not unique—it is perhaps the most unique that has been found, but it does not stand alone—merely demonstrates what can be done with radiant heating and proper orientation. Modern practice has just started to do the same kind of thing. The point is that we can do better than we are doing now. If the Romans achieved what they did with a very rudimentary technological apparatus, surely we can improve upon present methods of heating.

And I cannot entirely agree with Mr. Heubner in that the Romans were concerned with comfort for the few, only. The public baths were indeed public. The big ones accommodated thousands of people, and at a very low price of admission. They were subsidized by the State, evidently because they had become necessary social institutions for the general public.

The remark that Mr. Heubner quoted, in which I mentioned heating design being *forced into time-saving techniques*, is more a criticism of my own profession than of yours. I think that 80 percent of the time that heating engineers are given a job to do by an architect, it is a miracle that the building is comfortable, because the architect seems to have done his best to frustrate you. There should be a much closer liaison between the architect and the heating engineer. The architect should know a great deal more about heating than he does. I think that more and more the young architects are becoming aware of heating. They are also becoming aware of the importance of sunlight, in both heating and cooling, and of weather conditions, as well.

There are some pretty bad examples of building design, from a heating standpoint, I think. The buildings have been made comfortable through the efforts of the heating engineers, but the architects have certainly given them a hard time. The UN Building in New York is a case in point. If it had just been turned about 90 deg, it would have been a lot better building from the aspect of heating, as well as cooling. All the glass on the west face is a mistake.

To answer Mr. Linsenmeyer's question regarding temperatures, it is true that the climate around Rome is not severe. It is quite like ours, however, in that it goes through the same phases. Rome is about the latitude of Boston, and one finds the seasons progressing much the way they do here, but in not so extreme a manner. It is not so hot, or so cold.

In making the calculations for this building, hourly weather records for 1953 from the Rome Meteorological Bureau were gone through in detail. I took the lowest temperature in the winter of that year, which was 30 F. The average for the winter months would have been about 37 F. Wind speeds from the north were not very high; about 30 kilometers per hour was the maximum. The cold season there is from November through March, and January and February are the coldest months. December, March and November are increasingly mild, in that order. Northerly winds prevail during this November-March season, and there are intermittent winds from the southwest. These calculations, naturally, were concerned with heating, and not much attention was given to the summer temperatures.

Insofar as the interior temperatures were concerned, there is no way of knowing exactly what they were. Some ancient literary references leave no doubt that these bath buildings were very hot. There is one reference to some slaves who threw their master on the floor of the hot room and killed him. It is hard to tell whether the heat killed him or the beating killed him, but the writer did mention the fact that the floor was very hot. I assumed that, if Turkish baths are presently heated to about 110 or 120 F, these temperatures were more or less a maximum for the Roman system.

I made some experiments myself by standing, with bare feet, on a 100 F surface, and believe that a 100 F floor temperature is not too high. It was probably about average. Remember, this was not a living space; it was a sweat room.

It is true that if one stands on a 100 F floor for any length of time, he will be uncomfortable. But these people went to the baths to be uncomfortable, in a way. They wanted to sweat. With a 100 F floor temperature, the wall temperatures would have been between 90 and 95 F.

The air temperatures varied, depending upon the infiltration. In this particular building, with these open windows, I used 5 different conditions of infiltration: with doors open and doors closed, and with the oculus open and closed. The oculus was a round hole in the vault, used as a ventilating device. A shield-like metal fitting was hung under it, rigged on chains, and raised and lowered to vary the opening.

It is believed that under some conditions of infiltration the air temperature might have been 65 to 70 F. Most of the time it was probably around 80 F.

As far as domestic heating *is* concerned, just about everything is in the realm of guesswork. Most of the houses that were heated were heated only in one room, or in one suite of rooms. One finds houses with a warm room on the weather side—always on the weather, or northerly, side. This room will have at least a warm floor—a radiant floor. And perhaps the weather wall—the north, or west, or northwest wall—will be jacketed with flue tiles.

Mr. Heubner mentioned Fritz Kretzschmer, with whom the author has had a lot of correspondence. He is a German heating engineer who built some fires in a reconstructed bath building in the Saalburg, in southern Germany. The hot room had flue tiles on all 4 walls. He took temperature readings all over the room and found that this little bath room could achieve a 70 F air temperature with very little fuel and hardly any trouble at all.

With this experiment as a guide, one can only suppose that domestic heating could have achieved somewhat the same temperatures that are achieved today. But I think that those people did not need high air temperatures, partly because they dressed more warmly than we do, but principally because air temperature was not the governing factor. It was the mean radiant temperature that mattered.

So far as is known, they had no way of measuring temperature, particularly air temperature. It is to be presumed that they knew when they were comfortable, and that was all there was to it. I think that, with the clothing they wore, they would have been comfortable in a domestic room with an air temperature of 60 or 65 F, particularly when they had their feet on a warm floor and were sitting close to a radiant wall.

But all of this is highly conjectural. There are literally hundreds of examples of heating by hypocaust all over Europe. And, particularly in the north of Europe, there are a great many examples of domestic heating, alone, as distinct from bath heating. And the only one who has done anything about testing one of these is Kretzchmer. Some day, somebody will come along with some money and be able to rebuild one of these places, put some fire in it, and find out what really did happen.

1665

No. 1665

PROBE FOR THERMAL CONDUCTIVITY MEASUREMENT OF DRY AND MOIST MATERIALS

By W. WOODSIDE*, OTTAWA, CANADA

THE TRANSIENT probe or needle method for the measurement of thermal conductivity was originated by Stalhane and Pyk in 1931[1]. It was later developed by Van der Held and Van Drunen[2] who applied the method to the determination of the thermal conductivity of liquids.

The method utilizes a line heat source, *i.e.*, a straight wire through which constant electric current may be passed and a temperature sensitive device, *e.g.*, a thermocouple or resistance thermometer. The 2 elements are embedded alongside each other in the material under test. When the assembly is at a uniform and constant temperature a constant power input is supplied to the heater element and the rise in temperature is recorded during a short heating interval. The rate of rise of temperature is determined by the ability of the test material to conduct heat away from the line heater. The thermal conductivity may be calculated from the temperature-time record and the power input.

The method was further developed by Hooper and Lepper[3] and Hooper and Chang[4] who enclosed the 2 elements in a protective aluminum tube 5/16-in. in diameter and approximately 19 in. in length. This was the first actual probe. The reproducibility of this instrument was said to be 0.5 percent, and it was stated that the probe could measure the conductivity of both dry and wet materials with equal facility and precision. De Vries[5] also described a probe suitable for measurement of thermal conductivity of soils *in situ*, and now the method is widely used for soil resistivity surveys along proposed routes for buried electric cables.

Van der Held and Van Drunen[2], Lentz[6], Vos[7] and Blackwell[8] have investigated errors in the probe method. The theory of the method is based on the assumption of a perfect line heat source of infinitesimal diameter, and hence errors arise due

* Building Services Section, Division of Building Research, National Research Council.

[1] Exponent numerals refer to References.

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to the finite diameter of the probe. In an endeavour to reduce errors of this kind to a negligible level, D'Eustachio and Schreiner[9] constructed a probe approximately 4 in. long with an outside diameter of 0.030 in. Results for dry materials were in good agreement with measurements made by the conventional guarded hot plate method. The probe used in the present work is similar to that used by D'Eustachio and Schreiner, but with an even larger length to diameter ratio.

Joy[10] used a thin probe to determine conductivities of 3 insulating materials at a variety of moisture contents and at 2 mean temperatures (80 and 8 F). Both thermocouple and resistance thermometer-type probes were used. Joy obtained reproducibility of ±1 percent for tests on dry materials, and ±6 percent for tests on moist materials. He concluded that the probe method is suitable when the test specimen is uniformly wetted. The resistance thermometer probe had more integrating ability than the thermocouple type in the cases of non-uniformly wetted specimens.

Mann and Forsyth[11] described the construction of 4-in. long probes using hypodermic needle tubing (0.055-in. outside diameter). More recently, Lachenbruch[12] described a probe, containing a thermistor as the temperature-sensing element, for use in the determination of thermal conductivity of frozen soils *in situ*. Saare and Wenner[13] investigated the effect of the radial position of the thermocouple inside the probe.

EXPERIMENTAL PROCEDURE

A description of the probe and the auxiliary apparatus can be found in the Appendix.

The test material was placed, at the desired density, in an aluminum cylinder open at one end. The internal diameter of the cylinder was 6 in. and the internal height was 14.5 in. so that when the probe was inserted in the material at the axis of the cylinder, it was surrounded by 3 in. of material. The reference junction of the probe thermocouple was attached to the outer surface of the cylinder at mid-height. This assembly was placed centrally inside a larger cylinder of internal diameter 1 ft and height 2 ft. The annular space between the 2 cylinders was filled with granulated rock wool insulation. To prevent gain or loss of moisture from the test material, the top of the cylinder was covered with polythene film.

The system was allowed to attain thermal equilibrium before each test. This is indicated by zero emf in the thermocouple circuit.

To take full advantage of the entire width of the recorder chart, a portable precision potentiometer was used to provide a constant bias emf in the thermocouple circuit. Thus the recorder operates as zero-right or zero-left instead of zero-centre. The initial temperature rise from 0 to 2 minutes was not used to calculate thermal conductivity. Since the greater part of the absolute temperature rise occurs during these first 2 minutes, it is convenient to use the potentiometer emf source to bias out the initial rise, increase the preamplifier gain by a factor of 4, and re-adjust the bias emf so that at approximately 1.8 min the recorder pen is at zero. With this procedure, used only in the more recent tests, the temperature-rise curve between 2 and 10 min after the start of the test, *i.e.* the interval used for calculation of conductivity, is recorded on the full 9.5-in. width of the chart. This is possible since the calculation requires only the difference in temperature rises at two times, and not the absolute values of the rises. A typical recorder trace is shown in Fig. 1. Fig. 2 shows a recorder trace from a test in which the bias arrangement just described was used.

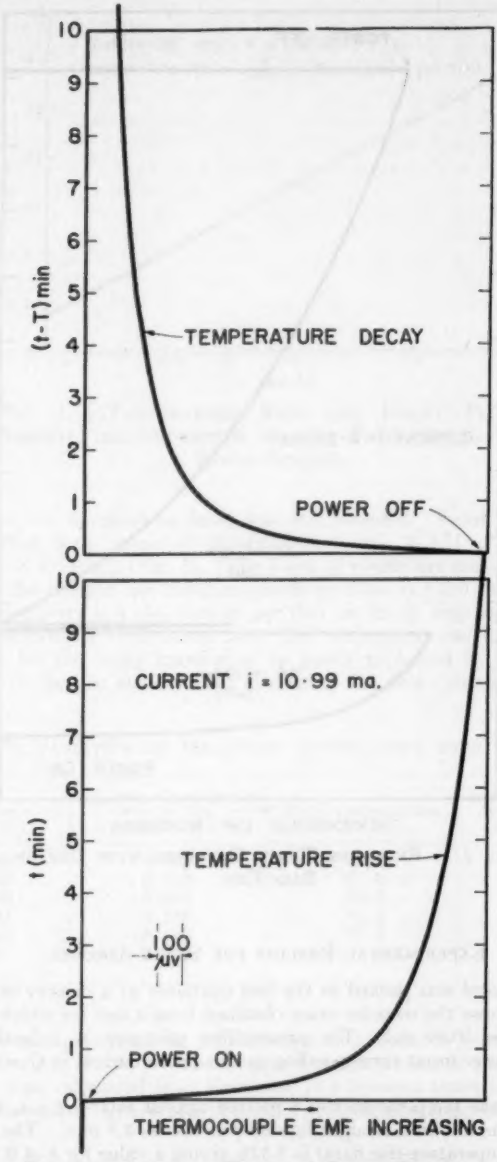

FIG. 1. . . . RECORDER TRACE OF PROBE TEMPERATURE RISE AND DECAY FOR TEST ON DRY SILICA AEROGEL

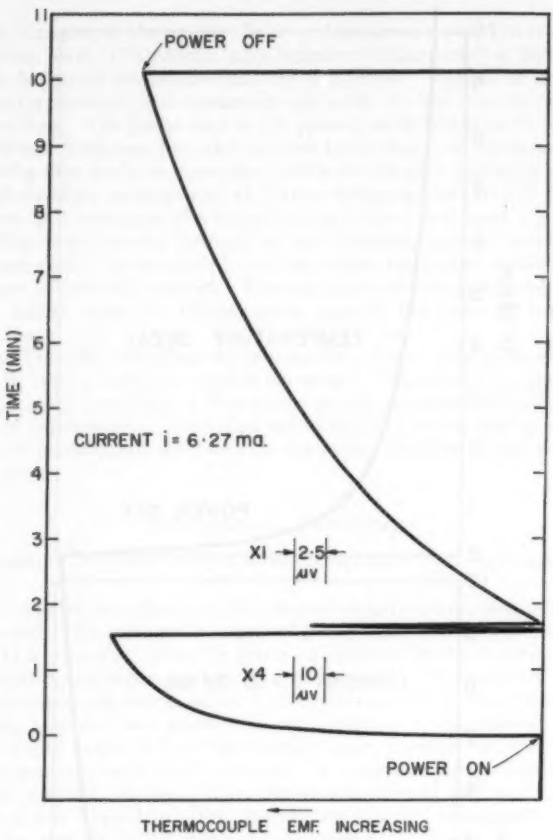

FIG. 2. . . .RECORDER TRACE OBTAINED WITH USE OF BIAS EMF

EXPERIMENTAL RESULTS FOR SILICA AEROGEL

Dry silica aerogel was packed in the test container at a density of 5.20 lb per cu ft. Fig. 1 shows the recorder trace obtained from a test for which the heating current was $i = 10.99$ *mA*. The preamplifier gain was × 5 (scale multiplier × 40). The power input corresponding to this probe current is $Q = 0.3260$ watt per ft.

Fig. 3 shows the temperature rise θ plotted against $ln(t + t_o)$, t_o being -0.25 min. The results fall on a straight line for t as low as 0.5 min. The slope of the straight line (temperature-rise data) is 5.325, giving a value for k of 0.199_8 Btu in. per (hr) (sq ft) (F deg). In this test the power was switched off at time $t = T =$

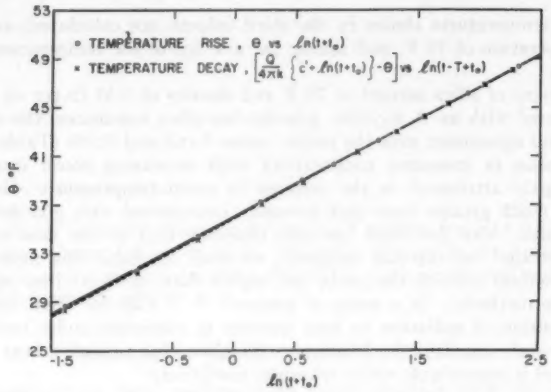

FIG. 3. . . .TEMPERATURE RISE AND DECAY PLOTTED AGAINST THE LOGARITHM OF THE TIME FOR TEST ON SILICA AEROGEL

10.5 min, and the temperature decay was also recorded. From the extrapolated temperature-rise data, values of $[(Q/4\pi k)[C' + ln(t + t_0)] - \theta]$ were plotted against $[ln(t - T + t_0)]$, (Fig. 3). The 2 sets of results are almost superimposed. The slope of the straight line (temperature-decay data) is 5.410, and the calculated thermal conductivity is 0.196_2 Btu in. per (hr) (sq ft) (F deg) agreeing very well with the conductivity determined from the temperature-rise data. The lower conductivity for the decay curve may be partly explained by the lower mean temperature during the measurement interval of the decay curve.

TABLE 1. . . .VARIATION OF MEASURED CONDUCTIVITY WITH POWER INPUT

POWER INPUT Q, WATT/FT	MEASURED CONDUCTIVITY BTU IN./HR, (SQ FT) (F)	MEAN TEMPERATURE, F	MAX. TEMP. GRADIENT AT PROBE SURFACE, F DEG PER IN.
0.00366	0.165	70.6	13.7
0.00632	0.169	71.0	23.5
0.01400	0.185	72.1	52.2
0.02927	0.186	74.2	109.1
0.05168	0.190	77.3	192.6
0.3260	0.199_5	129.4	1214.0

Tests were also performed on silica aerogel with different power inputs. The results are shown in Table 1. The measured conductivities in the second column of the table were calculated from Equation* A-4 between times of 2 and 10 min. The values shown are the averages of 7 values calculated during this time interval.

* Equation A-4 is developed in the Appendix, and is $k = 0.649_3[i^2 \log [(t_2 - 0.25)/(t_1 - 0.25)]]/(V_2 - V_1)$. In the Appendix also are Equations A-1, A-2, A-3 and A-5.

The mean temperatures shown in the third column are calculated, assuming an initial temperature of 70 F, and taking the average of the temperature rises at 2 and 10 min.

Conductivity of silica aerogel at 70 F and density of 5.57 lb per cu ft has also been measured with an 8- by 8-in. guarded hot-plate apparatus, the value being 0.167 in good agreement with the probe values 0.165 and 0.169 (Table 1).

The increase in measured conductivity with increasing power input may at least be partly attributed to the increase in mean temperature. However the increase is much greater than that normally experienced with guarded hot plate measurements. Van der Held has also observed that in the case of materials easily penetrated by thermal radiation, thermal conductivities measured by a transient method such as the probe are higher than conductivities measured by steady-state methods. In a series of papers[14, 15, 16] Van der Held has discussed the contribution of radiation to heat transfer in materials under both transient and steady-state conditions. It appears therefore that radiative heat transfer in silica aerogel is appreciable under transient conditions.

NOMENCLATURE

θ = temperature rise of probe, Fahrenheit.

Q = probe heat input per unit length, Btu per (hour) (foot).

r = radial distance from probe axis, feet.

r_o = probe radius, feet.

t = time from start of heating, hour.

T = duration of probe heating, hour.

t_o = probe time correction, hour.

τ = time for complete drying of volume element of specimen adjacent to probe, hour.

V = probe thermocouple emf corresponding to temperature rise, microvolts.

i = probe heating current, milliamperes.

p = water vapor pressure in pores of specimen, inches mercury.

k = thermal conductivity of test specimen, Btu per (hour) (foot) (Fahrenheit degree).

α = thermal diffusivity of test specimen, square feet per hour.

ρ = dry density of test specimen, pounds per cubic foot.

c = specific heat of test specimen, Btu per (pound) (Fahrenheit degree).

μ = water vapor permeability of test specimen pounds per (hour) (foot) (inches mercury).

m = initial moisture content of test specimen, fraction of dry weight.

Thermal diffusivity α of the test specimen may be estimated from the probe data, providing the effective radial distance, r, between the probe heater and thermocouple is known. It will be assumed that $r = 0.01$ in. (outside diameter of probe = 0.020 in.). From Fig. 3, it may be seen that when $ln(t + t_o) = 0$,

$$\theta = (Q/4\pi k)C' = 36.00.$$

Since

$$Q/4\pi k = 5.325,\ C' = 6.76,$$

but

$$C' = -0.577 - ln(r^2/4\alpha),$$

therefore

$$r^2/4\alpha = 6.51 \times 10^{-4},$$

and

$$\alpha = 0.016 \text{ sq ft per hr.}$$

Wilkes and Wood[17] give for silica aerogel at 147 F a value for α of 0.020 sq ft per hr, the density ρ being 4.0 lb per cu ft. Since the density in the probe tests was 5.2 lb per cu ft, and $\alpha = k/\rho c$ the probe diffusivity value agrees well with the value given by Wilkes and Wood.

EXPERIMENTAL RESULTS FOR MOIST SAWDUST

Several experiments were performed on moist sawdust with the probe, the moisture content and the probe power input being varied. Equation A-4 for the calculation of thermal conductivity from probe data is based upon heat conduction theory. It is known that heat transfer processes other than pure conduction occur in moist porous materials under a temperature gradient. However, it is assumed that the equation may be applied to the calculation of an effective thermal conductivity in the case of moist materials. In the tests on moist sawdust, the measured conductivity was found to vary during the time of the test. Fig. 4 shows a typical plot of conductivity *vs.* time after start of test. The conductivities shown on this graph are those calculated for times 0.5 to 1.0 min, 1.0 to 1.5 min, 2.0 to 3.0 min, 3.0 to 4.0 min, and so on. The conductivity apparently has a high initial value, quickly falling to a more or less constant value. This effect may be caused by the partial drying out of the test specimen in the immediate vicinity of the probe. Results are shown in Table 2. The conductivities tabulated are average values for the interval 2.0 to 10.0 min.

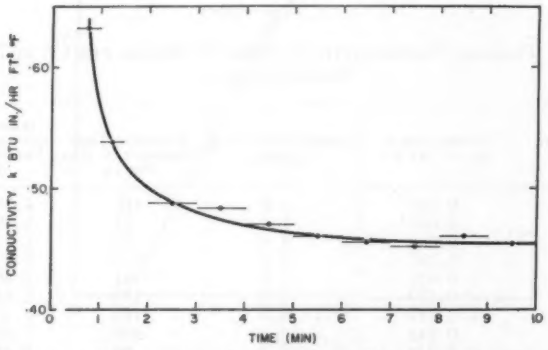

FIG. 4. . . .VARIATION OF MEASURED THERMAL CONDUCTIVITY WITH TIME DURING PROBE TEST ON MOIST SAWDUST

TABLE 2. . . .CONDUCTIVITY RESULTS FOR SAWDUST

DRY DENSITY, LB PER CU FT	MOISTURE CONTENT, % DRY WT	POWER INPUT, WATT PER FT	MEAN TEMP, F	MEASURED CONDUCTIVITY, BTU IN. PER (HR) (SQ FT) (F DEG)
13.00	3.14	0.1017	78.2	0.468
13.00	3.14	0.2425	83.5	0.471
13.00	3.14	0.5330	103.0	0.479
13.00	6.32	0.1186	79.0	0.505
13.00	6.32	0.2631	87.0	0.505
13.00	6.32	0.5838	105.0	0.508
13.84	12.8	0.1063	77.0	0.649
13.84	12.8	0.3223	85.4	0.635
13.84	12.8	0.7793	107.0	0.621

EXPERIMENTAL RESULTS FOR SNOW

For this series of tests, the test assembly was placed in a cold room[18], the probe power and thermocouple leads being taken out to an adjacent laboratory where the auxiliary equipment was situated. During the majority of these tests, the cold room was operated at a temperature of +15 F; the power input to the probe was limited in order that the temperature rise at the probe should not exceed 10 F deg.

Snow was first passed through a No. 6 sieve (opening 0.131 in.) and then placed in the test cylinder at a density of 17.83 lb per cu ft (0.286 gm per cc). Results of the tests on snow are shown in Table 3.

The measured conductivity varied appreciably from one test to the next, and also during the course of a single test. The measured conductivities reported in the last column of the table are the averages of 7 values calculated during the interval between 2 and 10 min from the beginning of each test. The initial temperatures given in the first column of the table are the air temperatures in the cold room during each test.

TABLE 3. . . .THERMAL CONDUCTIVITY OF SNOW (0.286 GM PER CC AT DIFFERENT TEMPERATURES)

INITIAL TEMP, F DEG	POWER INPUT, WATT PER FT	TEMPERATURE RISE, F DEG	MAXIMUM TEMP GRADIENT, F DEG PER IN.	MEASURED CONDUCTIVITY, BTU IN. PER (HR) (SQ FT) (F DEG)	
+15	0.242	6.0	110	1.51	
+15	0.054	1.3	27	1.39	
+15	0.054	1.3	27	1.54	Mean
+15	0.237	6.0	110	1.59	1.41
+13	0.329	7.5	161	1.40	
+13	0.270	6.0	185	1.01	
− 5	0.237	6.0	110	1.16	1.50
− 5	0.542	13.3	202	1.84	
−33	0.229	5.0	99	1.61	1.62
−33	0.231	5.0	99	1.63	

On averaging the conductivities obtained for the several tests at each temperature, the results indicate that the thermal conductivity of snow increases with decreasing temperature. Jakob[18] reports that the thermal conductivity of ice increases with decreasing temperature, and is 15.5 Btu in. per (hr) (sq ft) (F deg) at 32 F, 17.3 at −13 F, and 19.2 at −58 F. Thus, although the effective thermal conductivity of the air in the pore spaces in snow decreases with decreasing temperature, the net effect is an increase in the conductivity of snow with decreasing temperatures.

In the probe tests on moist sawdust the measured conductivity varied continuously with time during test. A variation in conductivity during test was also observed in the tests on snow; the variation was discontinuous, however. Fig. 5 shows the microvolt reading of the probe thermocouple plotted against log $(t + t_0)$ for a probe test on snow which was allowed to run for 25 min instead of the usual 10. For a normal material this graph would be a straight line, (*e.g.*, Fig. 3 for silica aerogel) the slope of which determines the thermal conductivity. In the

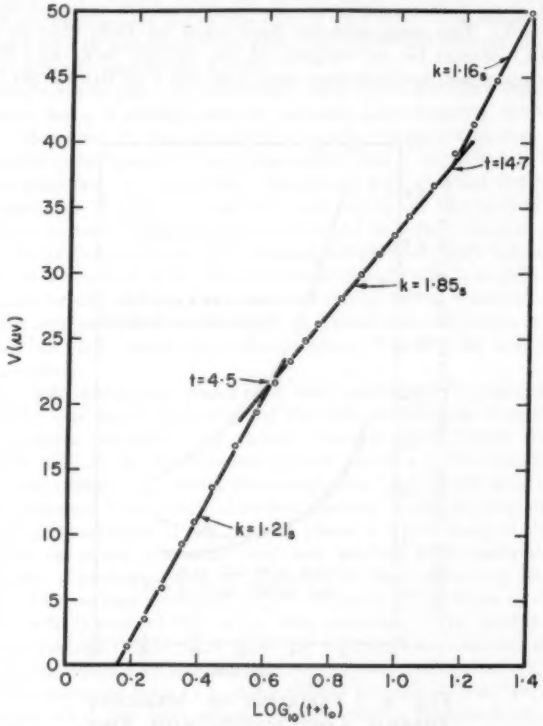

FIG. 5. . . .PROBE THERMOCOUPLE EMF PLOTTED AGAINST LOG (TIME) FOR TEST ON SNOW

present case, 3 straight lines each of a different slope, may be drawn through the test data. This corresponds to 3 different values of thermal conductivity: from 1.8 to 4.5 min $k = 1.21$; from 4.5 to 14.7 min $k = 1.85$; and from 14.7 to 25 min $k = 1.16$ Btu in. per (hr) (sq ft) (F deg). The times when a change in conductivity occurs are not the same from one test to the next and, as is evident from Fig. 5, the transition from one value of conductivity to another appears to be a sharp one.

Yosida[20] observed this same phenomenon during measurements of the thermal conductivity of snow, the method of measurement being very different from the one described here. The tests lasted about 30 min. Yosida attributed these changes in conductivity to changes in the snow structure brought about by the heat flow through the snow.

The thermal diffusivity α also may be calculated from the conductivity values shown in Table 3. The values of α range between 0.0132 and 0.0151 sq ft per hr.

Experimental Results for Moist Leda Clay

Results of probe measurements on dry Leda clay are reported by Woodside and de Bruyn[21]. Two measurements were made on Leda clay at a moisture content of 10.2 percent by dry weight, the dry density being 61.4 lb per cu ft. The average measured conductivities were 1.46 and 1.47 Btu in. per (hr) (sq ft)

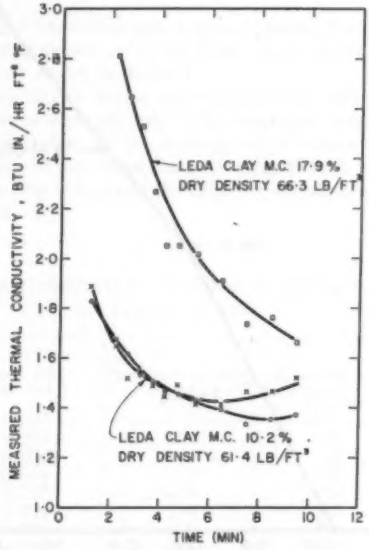

FIG. 6. . . . VARIATION OF MEASURED THERMAL CONDUCTIVITY WITH TIME DURING PROBE TESTS ON MOIST LEDA CLAY

(F deg). The variation of measured conductivity with time for these 2 measurements is shown in Fig. 6.

Five measurements were performed on Leda clay at a moisture content of 17.9 percent by dry weight, the dry density being 61.4 lb per cu ft. The average measured conductivities varied from 1.52 to 2.13 Btu in. per (hr) (sq ft) (F deg), the conductivity apparently increasing from one test to the next. An example of the very large variation in conductivity during the course of a single test is shown in Fig. 6.

DISCUSSION

Probe measurements on dry materials: The thermal conductivity probe method appears to be satisfactory for measurements on dry materials. The tests performed on dry silica aerogel gave results with excellent reproducibility, in good agreement with guarded hot plate data.

Several advantages of the probe method over the conventional guarded hot plate method are apparent. Testing time is of the order of 15 min compared with at least 24 hr for normal hot plate measurements. This does not mean, however, that 4 tests may be performed in 1 hr since the sample must be allowed to attain thermal equilibrium before the start of each test. The thermal diffusivity of the specimen may be estimated from probe data provided that the radial distance between thermocouple and heater is known.

Probe measurements may be performed with little auxiliary equipment, the basic necessities being a storage battery, portable potentiometer, stop watch and dewar flask. However, the measurements are more accurate and more convenient when a recording potentiometer with preamplifier and a variable bias emf source, such as those described, are available. Compared with guarded hot plate equipment, the apparatus is compact, portable and relatively inexpensive. In some instances, probe measurements might be performed on a test material *in situ*.

Tests may be performed with only a small temperature rise; for example, the temperature rise in several of the tests on silica aerogel was less than 1 F. This is an advantage for determining the thermal conductivity of materials at different mean temperatures, especially when the k *vs* temperature variation is non-linear. In the guarded hot plate, temperature differences of 30 to 40 F are normally applied across the specimens.

All the tests described in this paper have been performed on granular materials, and as the probe was easily inserted good thermal contact was obtained. In the case of non-granular materials, *e.g.*, wood, concrete, and fibrous insulation, it would be more difficult to obtain good thermal contact. Also many insulating materials are anisotropic, *e.g.*, wood, fiberboard, and many rock wool batt insulations. Such materials have vastly different thermal conductivities in directions parallel to and perpendicular to the *grain* or plane in which most of the fibers are oriented. Since the probe is a radial heat flow method, such materials introduce special problems. Specimens may be arranged so that tests may be performed first with the probe perpendicular (k_1) to the direction of the fibers of the material and, secondly, with it parallel (k_2) to the fiber direction. The conductivity k for heat flow perpendicular to the plane in which the fibers are oriented is then $k = (k_2)^2/k_1$. This procedure has been successfully applied to fiberboard by Joy[10].

The length to diameter ratio of a probe should be high so that the instrument approximates an ideal line heat source. Large probes require large specimens, however, and small probes tend to be comparable in diameter to the particles or cells of some porous materials.

Probe measurements on moist materials: In the determination of the thermal conductivity of moist materials, many problems arise. Most investigators in this field realize that moisture movement is not eliminated in probe measurements[10] but assume that the moisture redistribution during the short testing interval is negligible[3, 4, 5, 9]. For example, Hooper and Lepper[3] have stated that examination of the specimens following test indicated no substantial alteration in the moisture distribution. This is in marked contrast to measurements on moist materials with the guarded hot plate apparatus and other steady-state methods for measuring conductivity[21, 22, 23]. If a moisture redistribution should develop during a probe test, however, it would be difficult to detect since the greater part of the moisture content gradient would occur close to the probe surface.

The decrease in measured conductivity with time during the test interval, which was observed in the tests on moist sawdust and moist clay, indicates that (*a*) some drying out of the specimen close to the probe does occur; and/or (*b*) application of the heat conduction theory to the heat transfer mechanism which occurs in moist porous media is not justified.

Since the layers of test specimen close to the probe have the largest effect on the conductivity measured by the probe, any drying out of these layers would produce a large decrease in the measured conductivity. It would seem to be of interest, therefore, to calculate the rate of vapor flow induced by the temperature gradient. Such a calculation of necessity involves several assumptions and approximations.

The water vapor flow is given by the equation (Fick's law) $W = -\mu A\, dp/dr$, where μ is the vapor permeability of the specimen, A is the effective area for flow, and (dp/dr) is the vapor pressure gradient in the pore spaces of the specimen. Consider an element of test specimen in the shape of a concentric cylindrical shell surrounding the probe, of unit length and radius r. Both A and (dp/dr) will be evaluated at the outer surface of this shell. Thus $A = 2\pi r$.
Now

$$dp/dr = (d\theta/dr)(dp/d\theta)$$

But by Equation A-5 the steady-state temperature gradient is

$$d\theta/dr = -Q/2\pi rk$$

This is the overall temperature gradient in the material at radius r. The temperature gradient in the pore spaces of the material, however, may be many times greater than this overall gradient[24, 25], and it is the gradient in the pores which determines the corresponding vapor pressure gradient. For the present purpose, however, the overall gradient will be used. Therefore,

$$\begin{aligned} W &= \mu 2\pi r(Q/2\pi rk)(dp/d\theta) \\ &= \mu(Q/k)(dp/d\theta) \end{aligned}$$

The initial weight of moisture contained in the volume element is

$$M = \pi(r^2 - r_o^2)\rho m$$

where r_o is the probe radius, ρ is the dry density of the test specimen, and m is the initial moisture content of the specimen expressed as a fraction of the dry weight.

Thus the time τ required to dry out completely the volume element considered is

$$\begin{aligned} \tau &= M/W \\ &= [\pi(r^2 - r_o^2)\rho mk]/[\mu Q(dp/d\theta)] \end{aligned}$$

If a volume element of radius $2r_o$ is considered,

$$\tau = [3\pi r_o^3 \rho m k]/[\mu Q(dp/d\theta)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

This equation is now applied to the probe test on Leda clay with 10.2 percent moisture content by dry weight and 61.4 lb per cu ft dry density. The variation of measured conductivity with time for this test is shown in Fig. 6, and $r_o = 0.01$ in., $m = 0.102$, $\rho = 61.4$ lb per cu ft. The vapor permeability μ of this clay was measured by the dry cup test, yielding a value for μ of 33.9 perm. in. (grains in.) per (hr) (sq ft) (in. Hg). If the relative humidity within the pores of the clay is 100 percent at 85 F which is the mean temperature of the probe test, then

$$(dp/d\theta) = 0.0394, \text{ in. Hg per F deg}$$

Substituting these values in Equation 1, the value for τ is found to be 10.8 min. Thus the first layer of clay (of thickness equal to the probe radius) should lose all its moisture in approximately 11 min., and hence the measured thermal conductivity should decrease appreciably during the 10-min. test interval, as was observed.

Since the foregoing calculation leads to a conclusion contrary to the view now held by many workers in this field, the assumptions involved in the calculation are re-examined.

1. It was assumed that the steady-state temperature gradient is applied instantaneously and maintained during the test. The gradient at the probe surface is essentially the steady-state gradient as soon as the higher terms in the $I(x)$ series may be neglected, which is after 0.5 min for the probe described.

2. The calculation neglected the fact that as the volume element dries out, the relative humidity and hence the vapor pressure in the pores, decreases, thus reducing the flow.

3. It has been assumed that the only mechanism for moisture flow is thermally actuated vapor diffusion. Capillary forces, however, may tend to return liquid moisture to the drying regions.

4. Finally, as stated before, the temperature and vapor pressure gradients in the pore spaces may be several times higher than the corresponding overall gradients.

The effects due to 1, 2 and 3 tend to be wholly or partially compensated for by 4, so that the calculation leads to a result which is at least correct to an order of magnitude.

It may be concluded, therefore, that in some instances thermal conductivity probe measurements may produce moisture content gradients in the vicinity of the probe just as severe as those produced by steady-state hot plate measurements. In measuring thermal conductivity of moist materials, the probe sets up large temperature gradients for a short duration, whereas the guarded hot plate imposes relatively low gradients for a long duration. The ideal method, which would maintain as uniform a moisture distribution as possible, would appear to be one which is fast and uses small temperature gradients.

Equation 1 indicates that τ will be small, and hence the moisture content gradients set up will be large, when (*a*) probe radius is small; (*b*) power input to the probe is large; (*c*) mean temperature is high (*i.e.*, when $dp/d\theta$ is large); and (*d*) conductivity of the test specimen is small.

Factors (*a*), (*b*) and (*d*) also lead to high temperature gradients in the specimen. Thus it appears that moisture content redistribution in probe tests may be reduced by increasing the size of the probe and by using smaller power inputs.

From the point of view of probe design, the theory requires that the probe radius be as small as possible; for testing moist materials, however, it appears desirable that the probe radius be large. Thus the recent trend in probe development towards smaller probe diameters (large length to diameter ratios) appears to be in the wrong direction for the testing of moist materials.

CONCLUSIONS

1. It has been shown that the temperature gradients set up in the vicinity of a thin probe during test are exceedingly high, being of the order of hundreds of Fahrenheit degrees per inch.

2. The results of probe tests on dry silica aerogel have shown that both reproducibility and agreement with guarded hot plate results are good (better than 1 percent) when low probe power inputs are used.

3. Results of tests on moist sawdust have shown a steady decrease in measured conductivity during the 10-min test interval.

4. Results of tests on snow have shown a variation in measured thermal conductivity during test similar to that observed by Yosida who used an entirely different method of measurement.

5. In tests on moist clay, a steady decrease in measured conductivity during test was observed.

6. The probe method appears satisfactory for dry materials. The steady decrease in measured conductivity during test observed in the measurements on moist sawdust and moist clay suggests a steady drying out of the material in the vicinity of the probe.

7. A rough calculation indicates that in the test on 10 percent moisture content clay, a layer of clay equal in thickness to the probe radius, should completely dry out during the short 10-min test. Thus the moisture redistribution during test does not appear to be negligible as is commonly assumed, at least for tests using thin probes.

8. One way by which the temperature gradients and hence moisture redistribution in probe tests, may be reduced is to increase the probe radius, contrary to the present trend in probe development which is toward smaller and smaller probe sizes. Since errors due to the finite size of a probe may be corrected for, it would seem advantageous to use larger probes and avoid some of the difficulties in testing moist materials.

ACKNOWLEDGMENTS

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APPENDIX

THEORY OF THE PROBE METHOD

The transient line heat source theory on which the probe method is based is developed by Carslaw and Jaeger†. The temperature rise at a point in an infinite mass of material heated by a perfect line source is

$$\theta = \frac{Q}{2\pi k} I\left(\frac{r}{2\sqrt{(\alpha t)}}\right) \quad \text{(A-1)}$$

† H. S. Carslaw and J. C. Jaeger: *Conduction of Heat in Solids* (Clarendon Press, Oxford, 1947).

where

$$I(x) = C - \ln(x) + \frac{x^2}{2} - \frac{x^4}{8} + \cdots$$

where

C = Euler's constant (0.5772)

If $x = (r/2\sqrt{\alpha t})$ is sufficiently small, the terms of order x^2 and above in the $I(x)$ series may be neglected, and

$$\theta = \frac{Q}{2\pi k}\left[C - \ln\left(\frac{r}{2\sqrt{(\alpha t)}}\right)\right] \quad \text{(A-1a)}$$

Between times t_1 and t_2, the temperature rise $\Delta\theta$ is given by

$$\Delta\theta = \theta_2 - \theta_1 = \frac{Q}{4\pi k} \ln\left(\frac{t_2}{t_1}\right)$$

or

$$k = \frac{Q}{4\pi_1\Delta\theta_2} \ln\left(\frac{t_2}{t_1}\right) \quad \text{(A-2)}$$

The thermal conductivity may be evaluated therefore by measuring the temperature rise at 2 different times, and also the power input to the heater. An alternative method is to measure the slope $(Q/4\pi k)$ of the straight line obtained by plotting temperature rise θ *vs* $\ln(t)$.

Determination of Thermal Conductivity from the Temperature Decay Curve

Equation A-1a may be re-written as

$$\theta = \frac{Q}{2\pi k}\left[C - \ln\left(\frac{r}{2\sqrt{\alpha}}\right) + \tfrac{1}{2}\ln(t)\right]$$

$$= \frac{Q}{4\pi k}\left[C' + \ln(t)\right]$$

where

C' = a constant, independent of time.

When the power is switched off after a time $t = T$, the temperature decreases and eventually attains the initial value at the start of the experiment. Switching off the current at time T is equivalent to introducing at that time a heat sink of the same strength as the heat source, *i.e.*, a heat source of strength $(-Q)$. Therefore,

$$\theta = \frac{Q}{4\pi k}\left[C' + \ln(t)\right] \quad \text{for } t \leq T$$

and

$$\theta = \frac{Q}{4\pi k}\left[C' + \ln(t)\right] - \frac{Q}{4\pi k}\left[C' + \ln(t - T)\right] \quad \text{for } t > T.$$

The values of $Q/4\pi k[C' + \ln(t)] - \theta$ plotted against $\ln(t - T)$, therefore, must be a straight line, also with a slope of $Q/4\pi k$. The required values of $Q/4\pi k[C' + \ln(t)]$ for $t > T$ may be obtained by extrapolating the line obtained for $t \leq T$.

Thus the conductivity may be determined from both the temperature rise and the temperature decay curves.

SOURCES OF ERROR

There are many possible sources of error in the probe method other than the instrumentation errors in the measurement of power input, temperature rise and time. These are summarized below:

(*a*) Error due to dropping the higher terms in the $I(x)$ series. This error is small when r is small and t is large.

(*b*) Error due to finite length and finite diameter of heat source. Van der Held and van Drunen (Reference 2) showed that the presence of the instrument in the test material is equivalent to heat production before the start of measured time, *i.e.*, a time correction t_0 must be added to t_1 and t_2.

(*c*) Error due to the finite size of the sample. It may be shown that this error is negligible for a thin probe, if the specimen has a diameter of the order of 6 in., *i.e.*, the temperature rise at $r = 3$ in. after the normal length of heating time is negligible.

(*d*) Error due to contact resistance between probe and test material. D'Eustachio and Schreiner (Reference 9) observed that an air space between the probe and test material equal to half the probe diameter did not affect the results at low power inputs.

(*e*) Error due to variation in resistance of heater wire with temperature. If the heater wire is made of constantan or manganin, errors from this source are negligible.

(*f*) Errors due to inhomogeneity and anisotropy of test specimen. The probe measures a radial average conductivity. In the case of an anisotropic material which has different conductivities in mutually perpendicular directions, Scott†† has shown that the probe, when inserted perpendicular to the 2 principal directions, measures the geometric mean of the 2 principal conductivities.

DESCRIPTION OF THE PROBE AND AUXILIARY APPARATUS

The probe used in the experiments to be described was obtained commercially. It is 8.5 in. long and 0.020 in. in diameter (length to diameter ratio 425). The probe heater is a 0.012 in. OD bifilar coil of 0.001 in. constantan wire, with a resistance of 1867 ohms. The thermocouple is 0.0005 x 0.005 in. chromel-constantan. The heater and thermocouple are encased in a stainless steel tube of 0.020-in. OD and 0.002-in. wall thickness. The power and thermocouple leads are brought out of the tube through a plastic head.

Inserting the values for the resistance per unit length of heater, and the calibrated emf-temperature conversion factor for the thermocouple, Equation A-2 becomes

$$k = 0.649_2 \cdot \left[\frac{i^2 \log (t_2/t_1)}{(V_2 - V_1)}\right] \quad \text{(A-3)}$$

where

i = heating current, *ma*.

V_1, V_2 = thermocouple emf's in μV corresponding to the temperature rises at times t_1 and t_2 respectively.

The formula for the calculation of the power input to this probe is

$$Q = 2.69_8 \times 10^{-3} \times i^2, \text{ watts per foot.}$$

The circuit arrangement and auxiliary equipment used with the probe were designed to reduce instrumentation errors to a minimum. The circuit is shown in Fig. A-1. The power source is a d-c voltage regulator and controller assembly following the design of Dauphinee and Woods‡. The output voltage can be continuously varied between 0

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†† D. R. Scott: *Proceedings* (Physical Society, B 67, 1954, p. 731).

‡ T. M. Dauphinee and S. B. Woods: Low-level thermocouple amplifier and a temperature regulation system (*Review of Scientific Instruments*, Vol. 26, 1955, p. 693).

and 70v d-c, with a maximum load of 2.5 amp. The heating current i is determined by measuring the potential drop across a 0.1 Ω standard resistance in series with the heater, with a portable potentiometer of 160 mv range, readable to 1 μV. The emf corresponding to the temperature rise at the probe is amplified and then recorded on a single-point curve-drawing strip recording potentiometer with a range of −5, 0 and +5 mv, and a chart speed of 1 in. per min. The preamplifier has a maximum gain of 200, with a scale of −25, 0, +25 μV and scale multipliers of 1, 2, 4, 10, 20 and 40.

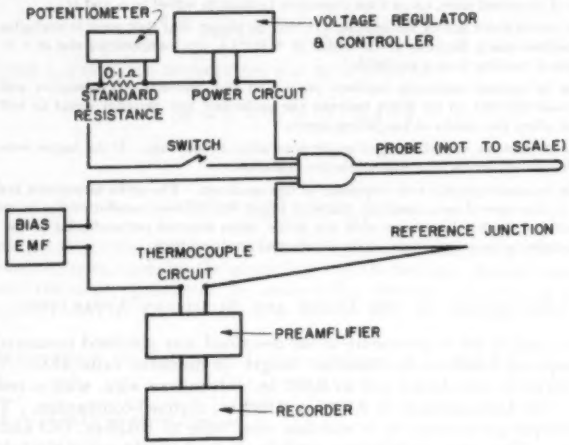

FIG. A-1....THERMAL CONDUCTIVITY PROBE CIRCUIT DIAGRAM

PROBE CHARACTERISTICS

Determination of t_o correction: Van der Held and van Drunen[2] have shown that the effect of a probe of finite dimensions is the same as that of an ideal line source, if a correction factor t_o is added to the time variable.

If Equation A-2 is differentiated and then inverted, it may be shown that

$$\frac{dt}{d\theta} = \frac{4\pi k}{Q} t_o = \frac{4\pi k}{Q} (t + t_o)$$

where

$t_e = t + t_o$ is the corrected time.

Thus when $dt/d\theta$ is zero,

$$t_e = t + t_o = 0, \quad \text{or} \quad t = -t_o.$$

Fig. A-2 shows a plot of dt/dV *vs* t, the results being obtained from a probe test on silica aerogel. From the graph, $t = -t_o = 0.25$ min, or $t_o = -0.25$ min for this probe.

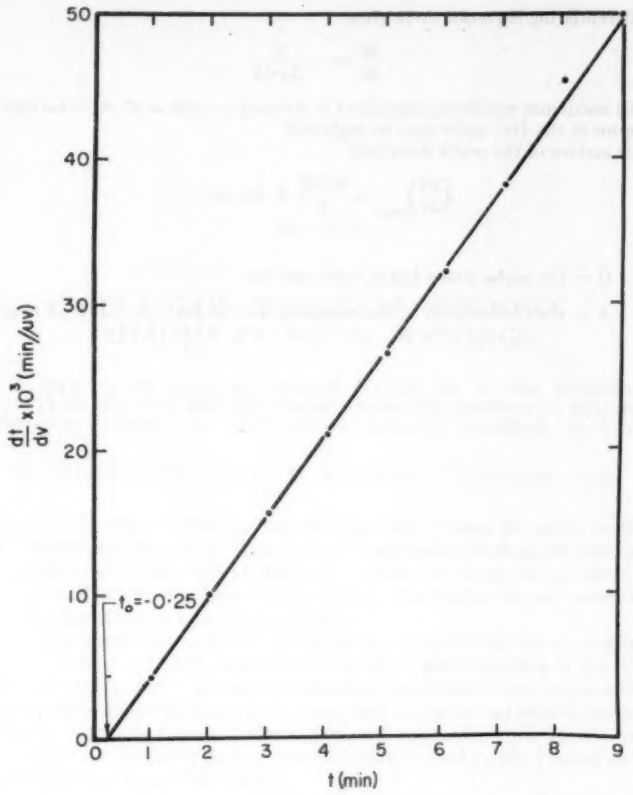

FIG. A-2. . . .GRAPHICAL DETERMINATION OF T_o CORRECTION FOR PROBE

Thus Equation A-3, when corrected, reads

$$k = 0.649_2 \times \frac{i^2 \log \left(\frac{t_2 - 0.25}{t_1 - 0.25} \right)}{(V_2 - V_1)} \quad \ldots\ldots\ldots \quad \text{(A-4)}$$

where

t_1 and t_2 = minutes.

Determination of maximum temperature gradient at probe surface: Under steady-state conditions when there is no change in the heat stored,

$$Q = -k(2\pi r)\left(\frac{d\theta}{dr}\right) \quad \text{or} \quad \left(\frac{d\theta}{dr}\right)_{max} = \frac{Q}{2\pi r k}$$

Also differentiating Equation A-1a gives

$$\frac{\partial \theta}{\partial r} = -\frac{Q}{(2\pi r)k}$$

Thus the maximum steady-state gradient is attained as soon as $x^2 = r^2/4\alpha t$ and higher-order terms in the $I(x)$ series may be neglected.

At the surface of the probe described,

$$\left(\frac{d\theta}{dr}\right)_{max} = \frac{652Q}{k}, \text{ F deg per in.} \quad \text{(A-5)}$$

where

Q = the probe power input, watts per foot.

k = the conductivity of the specimen, Btu in. per (hr) (sq ft) (F deg).

1666

No. 1666

HEAT GAIN THROUGH WINDOWS SHADED BY METAL AWNINGS

This paper is the result of research carried out by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Avenue, Cleveland 3, Ohio

By NECATI OZISIK* AND L. F. SCHUTRUM,** CLEVELAND, OHIO

THIS PAPER, which presents data on the effectiveness of metal awnings in reducing the transmission of solar heat through glass windows, is the result of research carried out at the ASHAE Research Laboratory under the guidance of the TAC on Heat Transfer through Fenestration†. An earlier paper presented the results of similar research on canvas awnings.

The awnings tested were selected at a meeting attended by the representatives of the metal awning industry, members of the TAC, and members of the ASHAE Research Laboratory staff. In making selections, the outside color, shape and venting characteristics of the awnings were recognized as significant factors influencing the heat gain. Because they provided a means of evaluating these factors and because they were considered typical of the more widely used types of metal awnings, the following were selected for testing:

1. Pan and cover type, sides closed, inside white, outside white.
2. Pan and cover type, sides closed, inside white, outside dark green.
3. Solid pan type, sides closed, inside white, outside white.
4. Horizontal louver type, sides open, inside white, outside white.
5. Horizontal retractable type††, sides closed, inside white, outside white.

White and dark green were chosen as the most commonly used colors to illustrate the effect of color. The solid pan type awning with closed ends was considered as representative of the type which afforded little or no venting, whereas the *louver* and *pan and cover* types permitted ample venting.

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* Research Engineer, ASHAE Research Laboratory.
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† Personnel: R. W. McKinley, *chairman;* R. C. Jordan, *vice chairman;* W. J. Arner, T. C. Carson, E. W. Conover, R. B. Crepps, W. B. Ewing, J. E. Frazier, H. F. Kingsbury, J. B. Leavy, Manny Levine, E. C. Miles, Bruno Morabito, F. W. Mowrey, D. J. Vild, H. B. Vincent, O. F. Wenzler, C. J. Youngblood, Jr.
Presented at the 65th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, January 1959.
†† Test was made only for spot-checking purposes.

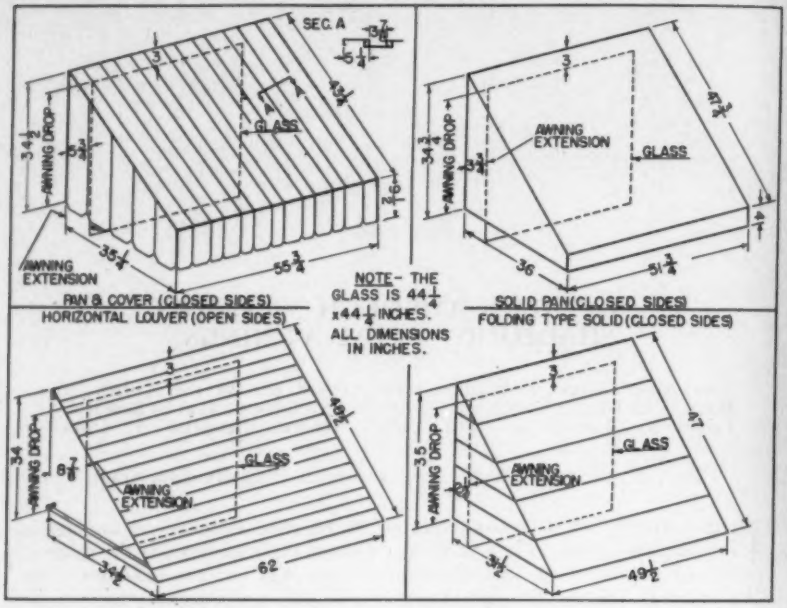

FIG. 1. . . . SHAPE AND SIZE OF METAL AWNINGS TESTED

The size and shape of the test awnings are given in Fig. 1. The solar reflectance (normal) of the awning material, as obtained from measurements made at the Laboratory using a pyrheliometer, and comparing it with a sample of known reflectance, was 0.30 for the dark green surface and 0.70 for white.

TEST APPARATUS

The heat gain was measured by means of the solar calorimeter essentially as described in references 1 and 2. The window of the calorimeter was 44¼ in. square, and consisted of ¼-in. thick regular plate glass.‡ The window had no setback. A general view of the calorimeter with the metal awning in position is shown in Fig. 2.

Measurements of direct, diffuse and ground reflected solar radiation were made with 3 Eppley thermoelectric type pyrheliometers. All temperatures were measured by copper-constantan thermocouples. Two 16-point electronic recorders provided continuous readings throughout the tests. Tests were made either with the calorimeter following the sun or for a fixed orientation.

The foreground immediately in front of the calorimeter was a dark-colored platform surrounded by a grass lawn.

‡ Transmittance for direct solar radiation (normal) 0.77.

FIG. 2. . . .VIEW OF THE SOLAR CALORIMETER

Heat Gain through Glass and Shade Combination: An analytical treatment for the heat gain through a window shaded by an awning was given in an earlier paper[1], and the same analysis was used in correlating the test data on metal awnings.

In the presence of solar radiation, the heat gain through a window is the sum of transmitted and convected-radiated gains. As explained in Reference 1, the fraction of solar radiation which passes directly through the glass is referred to as the transmitted heat gain; the heat dissipation from the warm glass into the room is known as the convection-radiation heat gain.

The test data for the transmitted component of the heat gain were correlated as a function of the dimensionless ratio (I_{GV}/I_{dV}), where I_{GV} and I_{dV} are respectively the intensity of ground reflected and total diffuse solar radiation falling upon a vertical wall having the same orientation as the window. Fig. 3 shows the transmittance□ for the awning-glass combinations as a function of (I_{GV}/I_{dV}) for the pan and cover, solid pan and horizontal louver (sides open) type awnings. The solid lines in this figure are the calculated values based on the analysis given in Appendix A of Reference 1.

Design data on the transmitted component of the total heat gain for the awning-glass combinations tested were calculated utilizing the relations given in Fig. 3. For the calculations, the solar radiation intensities were taken from ASHAE GUIDE for a clear atmosphere at 40 deg north latitude on August 1; and the ratio (I_{GV}/I_{dV})

[1] Exponent numerals refer to References.

□ Solar energy transmitted per unit area of the glass surface divided by the intensity of the total diffuse solar radiation falling on a vertical wall having the same orientation as the window.

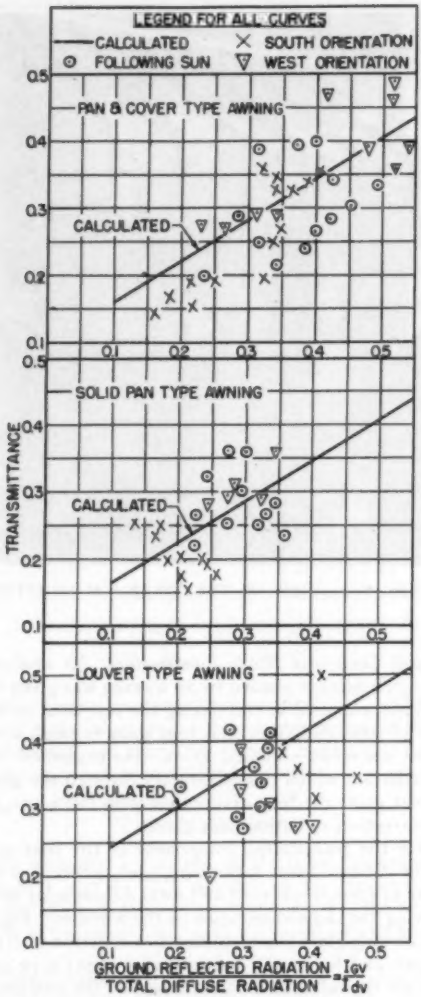

FIG. 3. . . .TRANSMITTANCE OF METAL AWNINGS WITH REGULAR PLATE GLASS

for each condition was determined as explained in Appendix B of Reference 1. These data are tabulated in Table 1A for east, south and west orientations and for various hours of the day.

TABLE 1. . . .DESIGN DATA ON HEAT GAINS THROUGH METAL AWNING-GLASS COMBINATION FOR 75 F INDOOR TEMPERATURE

TABLE 1. . . .DESIGN DATA ON HEAT GAINS THROUGH METAL AWNING-GLASS COMBINATION FOR 75 F INDOOR TEMPERATURE

Orientation	Sun Time	Outdoor Temp. F	Table – 1A			Table – 1B					Table – 1C					Table – 1D		
			Heat Gains[a] With Awnings													Total Heat Gain[a] Glass Only		
			Transmitted			Convection—Radiation					Total Gain					Type of Glass		
			Pan & Cover	Solid Pan	Louver	Pan & Cover		Solid Pan		Louver	Pan & Cover		Solid Pan		Louver	Reg. Plate (b)	Ord. Window (c)	Absor. Heat (d)
						White	Green	White	Green	White	White	Green	White	Green	White			
EAST	7AM	75	8+8[e]	7+8[e]	10	4	8	4	10	4	20	24	19	25	14	183	199	135
	8	77	10	9	12	6	11	8	14	6	16	21	17	23	18	192	211	142
	9	80	13	11	14	10	15	11	19	9	23	28	22	30	23	172	187	129
	10	83	13	11	14	12	19	14	22	12	25	32	25	33	26	124	137	94
	11	87	14	12	14	16	22	18	26	16	30	36	30	38	30	67	71	54
	12	90	13	11	12	18	24	20	26	18	31	37	31	37	30	32	33	27
SOUTH	8	77	8+1[e]	7	8+ 5[e]	5	8	6	10	4	14	17	13	17	17	19	20	14
	9	80	11	9	11+10[e]	8	13	10	16	8	19	24	19	25	29	44	48	34
	10	83	12	11	12+ 8[e]	11	18	14	21	12	23	30	25	32	32	73	78	57
	11	87	14	12	14	16	22	18	26	16	30	36	30	38	30	96	104	75
	12	90	14	13	15	20	26	22	30	20	34	40	35	43	35	107	115	84
	1PM	93	14	12	14	23	28	25	34	23	37	42	37	46	37	103	110	82
	2	94	12	11	12+ 8[e]	24	30	25	33	24	36	42	36	44	44	85	90	69
	3	95	11	9	11+10[e]	24	29	25	33	24	35	40	34	42	45	59	63	50
	4	94	8+1[e]	7	8+ 5[e]	22	26	23	28	21	31	35	30	35	34	36	37	32
WEST	12	90	13	11	12	18	24	20	26	18	31	37	31	37	30	32	33	27
	1PM	93	14	12	14	23	28	24	33	23	37	42	36	45	37	73	77	61
	2	94	13	11	14	24	31	26	35	24	37	44	37	46	38	136	147	107
	3	95	13	11	14	25	32	27	36	25	38	45	38	47	39	189	204	148
	4	94	10	9	12	24	30	25	33	24	34	40	34	42	36	210	228	164
	5	93	8+8[e]	7+8[e]	10	22	26	23	29	22	38	42	38	44	32	202	217	157

Note: Date obtained from a 44 in. square regular plate glass window facing a dark foreground and shaded by an awning having a drop of 70 percent. Values are calculated for a combined outside conductance of 3 Btu per (hr) (sq ft) (F deg).

[a] All heat gains are in Btu per (hour) (square foot).

[b] Regular plate glass, transmittance for direct-solar-radiation (normal) 0.77.

[c] Ordinary window glass, transmittance for direct-solar radiation (normal) 0.87.

[d] Heat absorbing glass, transmittance for direct-solar radiation (normal) 0.41.

[e] Addition due to direct sun on the glass.

The test data for the convection-radiation heat gains from the glass into the calorimeter, after being adjusted to a 75 F indoor temperature, are in good agreement with the calculated curve as shown in Fig. 4. (The calculated curve was obtained by the same treatment as that given in Appendix C, Reference 1.) As will be seen from this figure, the convection-radiation heat gain is a function of a term which is the sum of the outdoor air temperature (t_o), one-third the solar heat

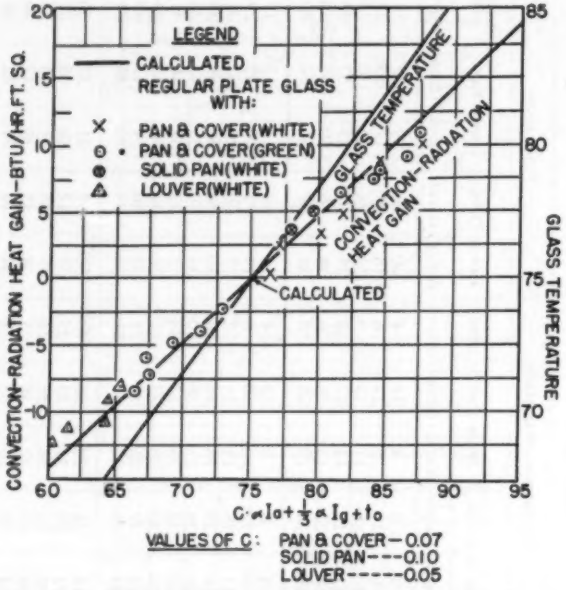

FIG. 4. . . .CONVECTION-RADIATION HEAT GAIN AND GLASS TEMPERATURE (FOR 75 F INDOOR TEMPERATURE)

absorbed per unit area of the glass (αI_g)*, and a fraction (determined experimentally) of the solar heat absorbed per unit area of the awning (αI_a)*. For wind velocities up to 5 mph, this fraction was approximately 0.10 for the solid pan-type awning, 0.07 for the pan- and cover-type awning, and 0.05 for the louver-type awning. For wind velocities above 5 mph, it was approximately 0.05 for all of the awnings tested.

The design data for the convection-radiation heat gains were calculated using the values from the curve in Fig. 4. Calculations were based on the same solar radiation intensities used in calculating the transmitted component. These data were tabulated in Table 1B for east, south and west orientations and for various hours of the day.

* aI_a, aI_g = solar energy absorbed by the awning and glass, respectively, Btu per (hour) (square foot).

The total heat gain through a window shaded by an awning is the sum of the transmitted and convected-radiated heat gains. The total gains for the awnings tested are given in Table 1C. These values are the sums of the components given in Tables 1A and 1B.

APPLICATION OF DATA

The data presented in Table 1 are strictly applicable only to the 44¼-in.-square regular plate glass test window in combination with the awnings tested, and facing a dark colored foreground. However, by making corrections described hereafter, the data in Table 1 may be applied to other size window-awning combinations and types of glass.

All the awnings tested had drops†† of approximately 70 percent. Provided that the awning drop remains the same, the variation in the width of window, within

TABLE 2. . . .SUNLIT AREA AT LOWER-SECTION OF THE GLASS AS PERCENTAGE OF THE ENTIRE GLASS SURFACE

ORIENTATION	SUN TIME	LOUVER TYPE (OPEN SIDES)			PAN & COVER AND SOLID PAN TYPES (CLOSED SIDES)		
		AWNING DROP			AWNING DROP		
		0.65	0.60	0.55	0.65	0.60	0.55
SOUTH	7 a.m. 5	—	—	—	—	—	—
	8 4 ↑	7	14	22	4	8	13
	9 ↓ 3	7	14	21	2	4	7
	10 2 p.m.	3	7	10	—	2	5
←EAST WEST→	7 a.m. 5 ↑	7	14	20	7	13	22
	8 4 ↑	—	—	4	—	7	15
	9 ↓ 3 p.m.	—	—	—	—	—	—

Note: Based on 3 ft x 5 ft window, awning extension 5 in. & 10 in. for conventional & louver type awnings respectively.

reasonable limits, does not appreciably alter the total heat gain given in Table 1C. If the awning drop is less than 70 percent, there may be additional sunlit glass areas in the lower section** of the glass which have not been accounted for in Table 1C. For awning drops other than 70 percent, the additional sunlit area, if any, in the lower section can be estimated from Table 2. Although this table is for a 3-ft × 5-ft window, it may be applied to windows of other dimensions without serious error. The heat gain through the additional sunlit area can be obtained from Table 1D.

The data in Table 1 are for a regular plate glass, but it can be applied with only a small error to awnings in combination with ordinary window glass or heat absorbing glass. The data in Table 1C are a few Btu low for the ordinary window glass and high for the heat absorbing glass.

†† Awning drop is defined as the fraction of the height of the window covered by the awning, see Fig. 2.
** Total glass height minus 1.4 times awning drop in feet.

The data in Table 1B and 1C are for an inside temperature of 75 F and for outside temperatures as indicated in Table 1 for different hours of the day. For temperature differentials other than those used in this table, the data may be corrected by adding or subtracting 1 Btu per (hr) (sq ft) for each degree of variation in temperature differential.

The data in Table 1C are for a dark foreground. For a light colored foreground, the values given in Table 1C should be increased by an amount approximately equal to the transmitted component of heat gain given in Table 1A (see Reference 1.)

Therefore, heat gain through any glass-awning combination having less than 70 percent drop for temperature differentials other than those on which Table 1C values are based and for a light colored foreground, can be determined in the following manner:

1. Heat gain through sunlit glass area in the lower section of the window is equal to the sunlit area from Table 2, multiplied by the appropriate value from Table 1D.

2. Heat gain through the remaining area of the glass is equal to the area of the remainder multiplied by the appropriate value from Table 1C.

3. For temperature differentials other than those on which Table 1C is based, data should be corrected by adding or subtracting an amount equal to the entire glass area multiplied by 1 Btu per hour for each degree of variation in temperature differential.

4. For a light colored foreground, increase in the heat gain is approximately equal to the entire glass area multiplied by the appropriate value from Table 1A.

Example: A southerly-oriented window having an ordinary window glass 3 ft wide and 5 ft high is shaded by a solid pan (sides closed) type white metal awning. The drop of the awning covers only 55 percent of the glass height. Calculate the total heat gain through the window at 3 p.m. for: 1. 75 F indoor temperature and a dark foreground; 2. 80 F indoor temperature and a dark foreground; 3. 80 F indoor temperature and a light colored foreground.

Solution: As the drop of the awning covers less than 70 percent of the entire glass height' the amount of sunlit area at the lower section of the glass should be determined. From Table 2, the sunlit area at 3:00 p.m. for solid pan type awning with 55 percent drop is about 7 percent of the entire glass surface. Therefore, the sunlit area $= 3 \times 5 \times 0.07 =$ 1 sq ft (approx.) and the shaded area $= 3 \times 5 - 1 = 14$ sq ft.

1. Total heat gain for a 75 F indoor temperature and a dark foreground: For the shaded area (from Table 1C, column 3): $14 \times 34 = 476$ Btu per hr; For the sunlit area (from Table 1D, column 2): $1 \times 63 = 63$ Btu per hr; Total heat gain through the entire glass surface: 539 Btu per hr.

2. Total heat gain for an 80 F indoor temperature and dark foreground: The air temperature differential for this example is $95 - 80 = 15$ F, whereas Table 1C value for 3 p.m. south is based on $95 - 75 = 20$ F. Allowing 1 Btu per (hr) (sq ft) correction per Fahrenheit deg difference in temperature differential, the reduction in heat gain is $15 \times 5 = 75$ Btu per hr. Hence, the total heat gain is $539 - 75 = 464$ Btu per hr.

3. Total heat gain for an 80 F indoor temperature and a light colored foreground: The additional heat gain due to the increase in the transmitted component of the heat gain (from Table 1A, column 2) is 9 Btu per (hr) (sq ft). Hence, the total heat gain is $464 + 9 \times 15 = 599$ Btu per hr.

DISCUSSION OF RESULTS

The above data on heat gain through metal awning and glass combinations are almost the same as the results obtained previously (Reference 1) with canvas awnings. The transmitted component of the heat gain differs only by a few Btu for

the two types. The difference is largely due to the variation in the size of awning openings, and the difference in the reflectance to solar radiation of the underside of the awnings.

The observed convection-radiation heat gain for metal awning and glass combinations correlates satisfactorily with the calculated curve shown in Fig. 4. It is interesting to note that the calculated curve for metal awnings is exactly the same as that shown in the previous paper for canvas awnings.

A metal awning having a high solar-absorbing surface on the outside, absorbs much solar radiation, which in turn increases the temperature of the awning, the temperature of air under the awning, and the convection-radiation heat gain into the room. Venting of such an awning tends to reduce the temperature of hot air under the awning, resulting in a little reduction in the convection-radiation heat gain. But with an awning having a low solar absorbing surface on the outside (*i.e.*,

TABLE 3. . . .HEAT EXCLUSION BY AWNINGS TESTED

TYPE OF AWNING	SOLAR HEAT EXCLUDED BY THE AWNING (PERCENTAGE)		
	EAST[a] ORIENTATION	SOUTH[b] ORIENTATION	WEST[c] ORIENTATION
Pan & Cover, outside white	81	58	75
Pan & Cover, outside dark green	77	51	70
Solid Pan, outside white	81	58	75
Solid Pan, outside dark green	76	48	69
Louver (sides open), outside white	81	51	75

[a] From 7 a.m. to 12 noon.
[b] From 8 a.m. to 4 p.m.
[c] From 12 noon to 5 p.m.

white), the awning temperature and the temperature of air under the awning remain about the same as the outdoor air temperature. The convection-radiation heat gain with a white awning is less than that with a vented awning having a high solar-absorbing outer surface. For a white awning venting is not important.

Table 3 was prepared to illustrate the effectiveness of the metal awnings tested in excluding solar heat over that period of a day during which the awnings stop the direct sun from falling upon the glass. (The data for this table are taken from Tables 1C and 1D.) This table gives the fraction of solar heat excluded by the awning for east, south and west exposures. The comparison of the results in this table shows that an awning with white color on the outside is more effective in excluding the solar heat.

CONCLUSIONS

1. The amount of solar heat excluded by a metal awning depends on the type and color of the awning, orientation of the window, and the time of the day. From the data presented in Tables 1A, B, C, D, and Table 2, the heat gain through a glass window shaded by a metal awning can be obtained for east, south and west exposures and for various hours of the day.

2. Over that period of a day during which an awning prevents the direct sun from falling upon the glass surface, the amount of solar heat excluded by the awning is 70 to 75 percent on a western exposure, 75 to 80 percent on an eastern exposure, and 50 to 60 percent on a southern exposure.

3. An awning having white color on the outside excludes more solar heat than the dark green one. Furthermore, venting of a white awning is not important.

ACKNOWLEDGMENT

The authors acknowledge with thanks the suggestions and guidance of the members of the Technical Advisory Committee on Heat Transfer through Fenestration, and the assistance of their colleagues of the ASHAE Research Laboratory staff.

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DISCUSSION

V. L. MILLER, Port Alleghany, Pa. (WRITTEN): ASHAE has made a number of studies to determine the effectiveness of shading devices in excluding solar heat. In all cases it has been shown that the devices can be made to effectively exclude significant factors of solar heat. It should also be pointed out that these shading devices also exclude a large fraction of the useful light. Only movable shades, carefully maintained in proper adjustment as the sun moves around, can do a good job of removing excess heat during periods when excess light is available. Perhaps ASHAE articles on shading data should include, or at least mention, this point of view to be fair to all types of fenestration, especially since shading devices remove so much light that supplementary room lighting may be necessary, and heat gain from such room lights may be greater than the heat gain excluded by the shading.

AUTHORS' CLOSURE (Mr. Ozisik): Thanks are due to Mr. Miller for pointing out the fact that shading devices also exclude a large fraction of the useful light. This matter was considered by the Technical Advisory Committee on Energy Flow Through Fenestration as a possible future research project.

1667

No. 1667

A NEW METHOD FOR CALCULATING RADIANT EXCHANGES

By BENJAMIN GEBHART*, ITHACA, N. Y.

MOST methods[1,2] for the determination of instantaneous radiant energy transfer rates between the gray surfaces of an enclosure (See, for example, Fig. 1) consider net transfer rates between pairs of surfaces. These net transfer rates are often ill-defined and awkward to use. Their calculation is difficult and involved in complicated enclosures. Further, the net transfer rate between two of the surfaces of an enclosure made up of many surfaces has no physical significance. It is not the rate of radiant energy loss or gain of either surface.

The present author has recently developed[3] a method of analysis which does not employ the net transfer method of energy analysis. Instead, attention is focused upon the total rate of radiant energy loss of a surface. The rate of loss is calculated by taking into account all of the radiant energy effects to which the surface is subject due to its surroundings. Only one energy quantity need be found for each surface and this quantity is equal to the instantaneous rate of energy loss by the surface due to radiation effects.

This rate of loss of a surface is defined as the rate of radiant energy emission by the surface minus the total rate of absorption of radiant energy at the surface. The absorption rate is calculated by taking into account all of the effects of the radiant surroundings, including all emissions and reflections. This instantaneous rate of loss has direct physical significance. It would be used in an energy balance for the surface just as the convection and conduction transfer rates are. In steady state, *i.e.*, the surface temperature not changing with time, the sum of the rates of radiant, convection, and conduction losses would be zero. In unsteady state this sum would be equal to the time rate of decrease of stored energy in the material of which the surface is made.

The rate of loss defined in this way is, therefore, similar to other quantities defined in heat transfer and is directly useful. For example, in a room heated by a radiant floor panel the necessary rate of energy supply by the embedded coils

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[1] Exponent numerals refer to References.

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would be the sum of the rate of loss by convection and the rate of loss by radiation as defined above. The convection loss may be calculated using the information on convection recently reported[4].

GENERAL ANALYSIS

Before applying the new method for several enclosure calculations, a brief account of the development is necessary. The relevant equations are presented and the significance of various quantities is discussed.

The rate of loss of a surface is defined as its emission rate minus the total rate of absorption of radiant energy from all sources. This rate of loss is affected by the emission rates of all the surfaces present, by their geometric relation with each other, and by their tendency to reflect radiant energy. Therefore, in order to find this rate of loss one must take into account the total radiant surroundings of the surface of interest and account for all paths by which radiant energy travels. The surface and its total surroundings may be considered an *enclosure* even though some of the enclosing surfaces have peculiar characteristics. For example, an opening transmits all radiant energy encountering it and emits none.

NOMENCLATURE

A_j = area of the jth surface, $j = 1, 2, 3, \ldots . n$.

B_{ij} = absorption factor, the fraction of the energy leaving A_i, uniformly, which is absorbed by A_j.

F_{ij} = angle factor from A_i to A_j.

n = total number of surfaces in an enclosure.

q_j = the net rate of energy loss from A_j due to thermal radiation processes.

T = absolute temperature.

W = emissive power per unit area and time = $\epsilon\sigma T^4$.

$a_{kl} = F_{kl}\rho_l$.

ϵ = emissivity.

ρ = reflectivity = $1 - \epsilon$ for opaque surfaces.

σ = 0.1713 x 10^{-8} Btu per (sq ft) (hr) (Rankine)^4.

τ = transmissivity of a window.

For an enclosure made up of n surfaces $A_1, A_2, \ldots . A_n$, an expression for the rate of loss of any one of them, called A_j, will be written. This quantity, called q_j, will be equal to the emission rate, W_jA_j, minus the absorption rates at A_j of a portion of the radiant emission from each of the n surfaces of the enclosure. If non-black surfaces are present these absorption rates must include direct incidence and all paths of reflections. A new quantity, similar to the angle factor, is called the absorption factor and is defined as the fraction of the total emission of one surface which is absorbed by another surface, including all paths whereby the energy may travel. Considering surface A_i which emits at a rate W_iA_i, B_{ij} is the absorption factor of A_j for the radiant emission of A_i. That is, B_{ij} is the fraction of the emission of A_i which is absorbed at A_j. Therefore, A_j absorbs energy from A_i at a rate of $B_{ij}W_iA_i$. The rate of loss equation is then written in these terms for A_j:

$$q_j = W_jA_j - B_{1j}W_1A_1 - B_{2j}W_2A_2 \cdots - B_{jj}W_jA_j \cdots - B_{nj}W_nA_n$$

$$= W_jA_j - \sum_{i=1}^{n} B_{ij}W_iA_i \quad \text{----------(1)}$$

Any enclosure surface may be taken as j. Note that some of W_jA_j may be reabsorbed at A_j; that is, B_{jj} is not necessarily zero.

Equation 1 is very similar to the one which would be written for an enclosure of black surfaces. The absorption factors would be replaced by angle factors and the emission rates would be those of black surfaces. This similarity provides the basis for an interpretation of the meaning of absorption factors. They are generalized angle factors between the surfaces of a non-black enclosure.

There are n absorption factors and they may be found by summing the absorption rates at A_j due to the emission rates and reflections at the other surfaces. For B_{1j}, W_1A_1 is emitted by A_1, $F_{1j}\epsilon_j$ is directly absorbed at A_j, and $F_{1j}\rho_j$ is reflected. Now the fraction of $F_{1j}\rho_j$ which is eventually absorbed at A_j is the same as the fraction of W_jA_j which is eventually absorbed at A_j, because the available paths are identical for the two quantities of radiant energy. Therefore, $F_{1j}\ \rho_j\ B_{jj}$ is absorbed by A_j. Similarly $F_{11}\ \rho_1$ is reflected from A_1 and $F_{11}\ \rho_1\ B_{1j}$ is eventually absorbed at A_j. B_{1j} is then written as:

$$B_{1j}=F_{1j}\epsilon_j+F_{11}\rho_1B_{1j}+F_{12}\rho_1B_{2j}+F_{13}\rho_3B_{3j}+\cdots F_{1n}\rho_nB_{nj}$$

Similar expressions are written for each surface and the results are rearranged in the n equations below. Note that $F_{k1}\ \rho_1$ is redesignated as α_{k1} for simplicity.

$$\left.\begin{aligned}
&(\alpha_{11}-1)B_{1j}+\alpha_{12}B_{2j}+\alpha_{13}B_{3j}+\cdots+\alpha_{1n}B_{nj}+F_{1j}\epsilon_j=0\\
&\alpha_{21}B_{1j}+(\alpha_{22}-1)B_{2j}+\alpha_{23}B_{3j}+\cdots+\alpha_{2n}B_{nj}+F_{2j}\epsilon_j=0\\
&\alpha_{31}B_{1j}+\alpha_{32}B_{2j}+(\alpha_{33}-1)B_{3j}+\cdots+\alpha_{3n}B_{nj}+F_{3j}\epsilon_j=0\\
&\quad\text{etc. to}\\
&\alpha_{n1}B_{1j}+\alpha_{n2}B_{2j}+\alpha_{n3}B_{3j}+\cdots(\alpha_{nn}-1)B_{nj}+F_{nj}\epsilon_j=0
\end{aligned}\right\}(2)$$

This set of equations may be solved for the n unknowns B_{1j}, B_{2j}, . . . B_{nj} as indicated in the original paper.[3] The foregoing set of equations is valid, with the same coefficients, for any choice of j, *i.e.*, 1, 2, ... n. Therefore, Equations 1 and 2 may be applied to any surface of the enclosure.

In an enclosure formed of n surfaces there are n^2 angle factors. However, as shown in Equations 3 and 4, there are necessary relations between the angle factors and one need not find all n^2 values from geometric considerations.

$$\sum_{j=1}^{n}F_{ij}=1 \qquad (3)$$

$$F_{ij}A_i=F_{ji}A_j \qquad (4)$$

Similarly, there are necessary relations of the same form between absorption factors[3] which simplify calculations.

$$\sum_{j=1}^{n}B_{ij}=1 \qquad (5)$$

$$B_{ij}\epsilon_iA_i=B_{ji}\epsilon_jA_j \qquad (6)$$

Equations 1, 2, 5 and 6 complete the basic information necessary for the use of the new technique.

Special Surfaces: All types of thermal radiation problems are included under the *enclosure* method just outlined However, some enclosures possess surfaces having very special properties in radiation exchange processes. Three such special kinds of surfaces are the window, the opening, and the adiabatic surface. Such surfaces

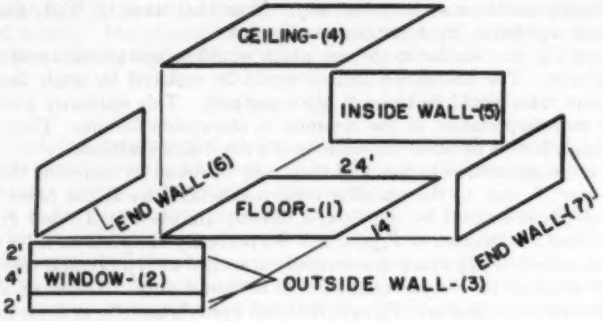

FIG. 1. . . .PANEL-HEATED ROOM CHOSEN FOR CALCULATIONS

are frequently encountered in room calculations. The characteristic of the opening is that it transmits all radiation encountering it and that radiation may enter the enclosure across it. The window absorbs, reflects, and transmits. The adiabatic surface has the characteristic of absorbing and emitting radiation in equal amounts.

If a window or opening transmits all radiation it is assigned a reflectivity of 0. If radiant energy enters the enclosure through a window or opening in sufficient quantities to be important in radiant exchange calculations, this energy quantity is accounted for by ascribing the necessary radiation characteristics to the equivalent window or opening surface.

The characteristic of an adiabatic surface, to absorb and emit at an equal rate, is simulated in this analysis by assuming that the surface absorbs and emits nothing. Its properties are taken as $\rho = 1.0$ and $\epsilon = 0$. That is, it is assumed to act as a perfect reflector. The present method offers a very convenient means of calculating the temperatures of adiabatic surfaces. This technique is demonstrated in the following calculations.

Radiation Transfer Rate Calculations: Calculations are presented below for radiation transfer rates and temperatures for various conditions in the typical 24 x 14 x 8 ft room diagrammed in Fig. 1. The 3 inside walls and ceiling are assumed adiabatic considering only radiation exchanges. The floor is a radiant panel. The outside wall and window exchange radiant energy with the floor.

TABLE 1. . . .GENERAL ENCLOSURE QUANTITIES

SURFACE	SURFACE CONDITION	AREA, SQ FT	REFLECTIVITY, ρ	EMISSIVITY, ϵ	ANGLE FACTOR TO: 1	2	3	4	5	6	7
1	80 F	336	0	1.00	0	0.087	0.088	0.454	0.175	0.098	0.098
2[a]	45 F	96	0.05	0.90	0.304	0	0	0.304	0.180	0.106	0.106
3	60 F	96	0.10	0.90	0.308	0	0	0.304	0.184	0.102	0.102
4	adia.	336	0.10	0.90	0.454	0.087	0.088	0	0.175	0.098	0.098
5	adia.	192	0.10	0.90	0.306	0.090	0.092	0.306	0	0.103	0.103
6	adia.	112	0.10	0.90	0.294	0.091	0.087	0.294	0.178	0	0.054
7	adia.	112	0.10	0.90	0.294	0.091	0.087	0.294	0.178	0.054	0

[a] The glass transmissivity and absorptivity for low temperature radiation are taken as $\tau = 0.05$ and $\alpha = 0.90$.

The surface properties and angle factors are listed in Table 1. The floor is assumed covered with a fabric which renders it essentially black.

Four different calculations are shown for this room. The floor radiant energy loss rate is calculated grouping the 4 adiabatic surfaces into one as a first approximation. The calculation is then repeated grouping only the 2 end walls into a single surface and results are compared. Next, the radiant losses involving the window are calculated. Finally, the average adiabatic surface temperature is calculated.

FLOOR LOSS CALCULATIONS

(*a*) With the 4 adiabatic surfaces grouped into one, hereafter called A_a, there is a 4-surface enclosure with characteristics as listed in Table 2. There are 4 absorption factors and four equations between them of the form of Equations 2.

$$(\alpha_{11}-1)\beta_{11}+\alpha_{12}\beta_{21}+\alpha_{13}\beta_{31}+\alpha_{1a}\beta_{a1}+F_{11}\epsilon_1=0$$
$$\alpha_{21}\beta_{11}+(\alpha_{22}-1)\beta_{21}+\alpha_{23}\beta_{31}+\alpha_{2a}\beta_{a1}+F_{21}\epsilon_1=0$$
$$\alpha_{31}\beta_{11}+\alpha_{32}\beta_{21}+(\alpha_{33}-1)\beta_{31}+\alpha_{3a}\beta_{a1}+F_{31}\epsilon_1=0$$
$$\alpha_{a1}\beta_{11}+\alpha_{a2}\beta_{21}+\alpha_{a3}\beta_{31}+(\alpha_{aa}-1)\beta_{a1}+F_{a1}\epsilon_1=0$$

The α_{k1} values and constants are computed from the F, ρ and ϵ values and are listed in Table 2. Note that since $\rho_a = 1.0$, $\epsilon_a = 0$ and $W_a = 0$, only the first 3 absorption factors need be computed. The equations are:

$$-B_{11} + 0.0044B_{21} + 0.0088B_{31} + 0.825B_{a1} + 0 = 0$$
$$0 - B_{21} + 0\,B_{31} + 0.696B_{a1} + 0.304 = 0$$
$$0 + 0\,B_{21} - B_{31} + 0.692B_{a1} + 0.308 = 0$$
$$0 + 0.0044B_{21} + B_{31} + (1 - 0.454)B_{a1} + 0.369 = 0$$

From which $B_{11} = 0.5836$, $B_{21} = 0.7875$, and $B_{31} = 0.7890$. The rate of loss from the floor is found as:

$$q_1 = W_1A_1 - B_{11}W_1A_1 - B_{21}W_2A_2 - B_{31}W_3A_3 - B_{a1}W_aA_a$$
$$= 10^3(48.94 - 28.56 - 7.58 - 8.54) = 4260 \text{ Btu per hr}$$

TABLE 2. . . .ENCLOSURE QUANTITIES WITH THE 4 ADIABATIC SURFACES COMBINED

SURFACE	SURFACE CONDITION	AREA, SQ FT	REFLECTIVITY, ρ	EMISSIVITY, ϵ	ANGLE FACTOR TO: 1	2	3	a
1	80 F	336	0	1.0	0	0.087	0.088	0.825
2	45 F	96	0.05	0.90	0.304	0	0	0.696
3	60 F	96	0.10	0.90	0.308	0	0	0.692
a	adia.	752	1.0	0	0.369	0.089	0.088	0.454

SURFACE	T(ABS)	$W = \epsilon\sigma T^4$	$WA \times 10^{-3}$	F_{11}	α_{k1}[a] 1	2	3	a
1	540	145.66	48.941	0	0	0.0044	0.0088	0.825
2	505	100.27	9.626	0.304	0	0	0	0.696
3	520	112.72	10.821	0.308	0	0	0	0.692
a	. . .	0	0	0.369	0	0.0044	0.0088	0.454

[a] To be used in Equations 2.

TABLE 3. . . .ENCLOSURE QUANTITIES WITH ONLY TWO ADIABATIC SURFACES COMBINED

SURFACE	SURFACE CONDITION	AREA, SQ FT	REFLECTIVITY ρ	EMISSIVITY ϵ	ANGLE FACTOR TO: 1	2	3	4	5	6
1	80 F	336	0	1.00	0	0.087	0.88	0.454	0.175	0.196
2	45 F	96	0.5	0.90	0.304	0	0	0.304	0.180	0.212
3	60 F	96	0.10	0.90	0.303	0	0	0.304	0.184	0.204
4	adia.	336	1.0	0	0.454	0.087	0.088	0	0.175	0.196
5	adia.	192	1.0	0	0.306	0.090	0.092	0.306	0	0.206
6	adia.	224	1.0	0	0.294	0.091	0.088	0.294	0.179	0.054

SURFACE	T(ABS)	$W = \epsilon\sigma T^4$	$WA \times 10^{-3}$	$F_{11}\epsilon_1$	α_{k1} 1	2	3	4	5	6
1	540	145.66	48.941	0	0	0.0044	0.0088	0.454	0.175	0.196
2	505	100.27	9.626	0.304	0	0	0	0.304	0.180	0.212
3	520	112.72	10.821	0.308	0	0	0	0.304	0.184	0.204
4	. . .	0	0	0.454	0	0.0044	0.0088	0	0.175	0.196
5	. . .	0	0	0.306	0	0.0045	0.0092	0.306	0	0.206
6	. . .	0	0	0.294	0	0.0045	0.0088	0.294	0.179	0.054

(*b*) If only the end walls are grouped into one surface, called A_b, a 6-surface enclosure results. Table 3 summarizes the properties, angle factors, and α_{k1} values. Six equations result for the 6 absorption factors B_{11}, B_{21}, B_{31}, B_{41}, B_{51} B_{b1}. However, since ϵ_4, ϵ_5 and ϵ_b are zero the last three factors need not be found.

$$-B_{11}+0.0044B_{21}+0.0088B_{31}+0.454B_{41}+0.175B_{51}+0.196B_{b1}+0=0$$

$$0-B_{21}+0+0.304B_{41}+0.180B_{51}+0.212B_{b1}+0.304=0$$

$$0+0-B_{31}+0.304B_{41}+0.184B_{51}+0.204B_{b1}+0.308=0$$

$$0+0.0044B_{21}+0.0088B_{31}-B_{41}+0.175B_{51}+0.196B_{b1}+0.454=0$$

$$0+0.0045B_{21}+0.0092B_{31}+0.306B_{41}-B_{51}+0.206B_{b1}+0.306=0$$

$$0+0.0045B_{21}+0.0088B_{31}+0.294B_{41}+0.179B_{51}-0.946B_{b1}+0.294=0$$

Solution of these equations yields $B_{11} = 0.5838$, $B_{21} = 0.7845$ and $B_{31} = 0.7858$. The rate of radiant energy loss is computed as before.

$$q_1 = W_1A_1 - B_{11}W_1A_1 - B_{21}W_2A_2 - B_{31}W_3A_3$$
$$= 10^3(48.94 - 28.57 - 7.55 - 8.50) = 4316 \text{ Btu per hr}$$

Comparison of the answers of calculations (*a*) and (*b*) show that combining all of the adiabatic surfaces into one produced only something over 1 percent error in the energy loss of the floor panel. Such uncertainties are certainly unimportant and the combination of several areas having the same properties is usually permissible. The error may become more important, however, for surfaces having lower emissivities.

RADIATION CALCULATIONS FOR A WINDOW

The rate of radiant energy loss from the inside surface of the window (to the room surfaces) of Fig. 1 is next calculated, grouping all adiabatic surfaces into one

area. The rate of radiant energy transmission from the room to the outside through the window is also computed.

The absorption factors B_{12}, B_{22}, and B_{32} are found from the following equations as 0.1978, 0.1006, and 0.1000 respectively. The α_{k1} values are the same as used for computing the floor loss. They are listed in Table 2. The rate of loss from the inside window surface, q_2, is then found.

$$(a_{11} - 1)B_{12} + a_{13}B_{32} + a_{13}B_{32} + a_{1a}B_{a2} + F_{12}\epsilon_2 = 0$$

$$a_{21}B_{12} + (a_{22} - 1)B_{22} + a_{23}B_{32} + a_{2a}B_{a2} + F_{22}\epsilon_2 = 0$$

$$a_{31}B_{12} + a_{32}B_{22} + (a_{33} - 1)B_{32} + a_{3a}B_{a2} + F_{32}\epsilon_2 = 0$$

$$a_{a1}B_{12} + a_{a2}B_{22} + a_{a3}B_{32} + (a_{aa} - 1)B_{a2} + F_{a2}\epsilon_2 = 0$$

$$q_2 = W_2A_2 - B_{12}W_1A_1 - B_{22}W_2A_2 - B_{32}W_3A_3$$

$$= 10^3(9.63 - 9.68 - 0.97 - 1.08) = -2100 \text{ Btu per hr}$$

The minus sign means that the window gains radiant energy from the surfaces inside the room.

The amount of the radiation of the enclosure transmitted through the glass to the outside is calculated by observing that for every Btu incident upon the window, 0.90 Btu is absorbed and 0.05 Btu is transmitted. Now, the amount $B_{12}W_1A_1 + B_{22}W_2A_2 + B_{32}W_3A_3$ is absorbed by the window. Therefore, $1/\rho_2$ times this absorbed amount is incident and τ_2/ρ_2 is transmitted. The transmitted amount, q_T, is therefore,

$$q_T = (B_{12}W_1A_1 + B_{22}W_2A_2 + B_{32}W_3A_3)\,\tau_2/\rho_2$$

$$= (9.68 + 0.97 + 1.08) \times 10^3 \times 0.05/0.90 = 650 \text{ Btu per hr}$$

This is seen to be a small quantity. For an opening, having $\tau = 1.0$, the amount would be much greater.

Another convenient aspect of this method of analysis is that the sum of all the surface heat losses and transmissions as defined here is equal to zero. Therefore, q_3 may be found from the results of the previous calculations directly.

$$q_1 + q_2 + q_3 - q_T = 0$$

$$q_3 = q_T - q_1 - q_2 = 650 - 4260 + 2100 = -1510 \text{ Btu per hr}$$

AVERAGE TEMPERATURES OF ADIABATIC SURFACES

In the preceding calculations the characteristic of an adiabatic surface, to absorb and emit at an equal rate, was simulated by assigning to the surface the fictitious properties $\rho_a = 1.0$ and $\epsilon_a = 0$. No absorption and no emission certainly guarantees that the surface will be adiabatic. However, in order to calculate adiabatic surface temperatures these surface properties may not be used. The surface temperature results from the fact that the emission rate must equal the rate of total energy absorption. The actual surface properties are used.

The adiabatic condition is fulfilled by requiring that the rate of loss, q_a, is zero. Therefore, the expression for q_a is found in terms of the absorption factors, the known emission rates, and the areas. W_a is then found and from this T_a is computed.

The absorption factors B_{1a}, B_{2a}, B_{3a} and B_{aa} are found from the following equations. Some α_{k1} values are different from those used in the preceding calculations because of the different surface properties now assigned to A_a. The constant terms

TABLE 4. . . .ENCLOSURE QUANTITIES (ACTUAL PROPERTIES FOR ALL SURFACES)

SURFACE	REFLECTIVITY, ρ	EMISSIVITY, ϵ	$WA \times 10^3$	$F_{1a}\epsilon_a$	σ_{k1}			
					1	2	3	a
1	0	1.0	48.941	0.742	0	0.0044	0.0088	0.0825
2	0.05	0.90	9.626	0.626	0	0	0	0.0696
3	0.10	0.90	10.821	0.623	0	0	0	0.0692
a	0.10	0.90	Unknown	0.404	0	0.0044	0.0088	0.0454

in the equations are now $F_{1a}\epsilon_a$, $F_{2a}\epsilon_a$, $F_{3a}\epsilon_a$, and $F_{aa}\epsilon_a$. This information is summarized in Table 4.

$$-B_{1a} + 0.0044B_{2a} + 0.0088B_{3a} + 0.0825B_{aa} + 0.742 = 0$$

$$0 - B_{2a} + 0 + 0.0696B_{aa} + 0.626 = 0$$

$$0 + 0 - B_{3a} + 0.0692B_{aa} + 0.623 = 0$$

$$0 + 0.0044B_{2a} + 0.0088B_{3a} + 0.955B_{aa} + 0.404 = 0$$

The absorption factors are found as $B_{1a} = 0.7865$, $B_{2a} = 0.6561$, $B_{3a} = 0.6529$, and $B_{aa} = 0.4321$. The expression for q_a is written and solved for W_a. T_a is then computed.

$$q_a = 0 = W_a A_a - B_{aa} W_a A_a - B_{1a} W_1 A_1 - B_{2a} W_2 A_2 - B_{3a} W_3 A_3$$

$$W_a = \frac{B_{1a} W_1 A_1 + B_{2a} W_2 A_2 + B_{3a} W_3 A_3}{A_a (1 - B_{aa})} = 118.7 \text{ Btu/(hr)(sq ft)}$$

$$T_a^4 = \frac{W_a}{\epsilon_a \sigma} = 769.8 \times 10^8, \quad T_a = 527\text{R or } 67\text{F}$$

This temperature is an estimate of the average temperature of the surfaces, assuming that they are adiabatic from the radiation point of view.

If conduction through the surface and convection from the surface to the room air are present, the surface will be adiabatic from the radiation point of view only if the conduction and convection effects balance each other. In general these effects do not balance and the surface temperature is estimated from the surface energy balance. This balance includes q_a for radiation, q_k for conduction, and q_c for convection. The equation is

$$q_a + q_k + q_c = 0$$

$$W_a A_a - \sum_{i=1}^{n} B_{ia} W_i A_i + \frac{k A_a}{\Delta x}(T_a - T_0) + h_c A_a (T_a - T_r) = 0 \quad \cdots (7)$$

where

$T_a - T_0$ = conduction absolute temperature difference across the wall.

T_r = absolute room air temperature.

k = thermal conductivity of the wall material.

h_c = average convection coefficient for the adiabatic surface.

The solution of this equation for T_a is more difficult; therefore, an approximate technique would be employed.

SUMMARY AND RECOMMENDATIONS

A new method of thermal radiation analysis for non-black surfaces is demonstrated by calculating various radiant energy transfer rates in a room. The analysis is direct and consistent because of the way in which basic quantities are defined and is similar to the approach used for black surfaces. The same basic technique is used in all calculations with only minor alterations.

In the calculations, as in any calculations which account for many processes simultaneously, the numerical operations are lengthy. The involved calculations arise in the solution of the equations for the absorption factors. For a many-surface enclosure, this is necessarily a machine calculation.

Since the major inconvenience in the use of the method is in connection with the calculation of absorption factors, the question arises as to whether or not the values of the absorption factors could be presented graphically. This is the standard practice for the similar quantity, angle factor. Perhaps it is also possible and practical to tabulate or graph complete solutions for the heat loss rate and to present design information directly.

A preliminary study indicates that much can be done along these lines for room-like enclosures. Room surface emissivitives, angle factors, and temperatures vary over relatively narrow ranges and many simplifications and approximations are possible. Therefore, if this method is adjudged of value in this application, it is recommended that studies be undertaken, first, to determine what is practical in the form of general solutions, and second, to make the calculations and to organize the results in a form applicable to design. In any case, it is considered desirable, as a first step, to present absorption factors graphically in order to facilitate various types of calculations such as those for adiabatic surface temperatures.

ACKNOWLEDGMENT

The author wishes to express his thanks to C. O. Mackey for numerous helpful discussions and suggestions.

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DISCUSSION

K. R. SOLVASON†, Ottawa, Ontario, Canada (WRITTEN): The author is to be complimented for the presentation of a method for the calculation of radiant exchanges. Practical application of the method outlined would, however, appear to be limited to calculations of the radiant component of the exchange between surfaces of known temperature, or for a complete analysis of the limited number of cases where the convective exchange may be disregarded. In most air-conditioning heat exchange problems the radiant and convective components must be treated simultaneously because of their interrelation. Even in the example given by the author the calculated radiant emission

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from the floor slab can be correct only if there is no convective transfer at the adiabatic walls, since any convective transfer will change the adiabatic wall surface temperatures and hence, the emission from the slab.

A rigorous mathematical treatment of the exchange in a simple room would, in most cases, be impractical because of the many lengthy calculations required, since in addition to treating radiation effects proportional to the 4th power of temperatures, convective effects proportional to some power of the air to surface temperature differences must be considered simultaneously. Such a treatment could no doubt be performed using modern computers or analog techniques, otherwise, it would seem more practical to employ simple surface coefficients adjusted appropriately for the expected radiant and convective components.

Consider for example an air-conditioning unit to maintain a constant air temperature in a simple room with one wall and window exposed to periodically varying outside temperature and solar radiation, the other walls, floor and ceiling exposed to constant temperature air. Heat enters the room by conduction through the exposed wall and by radiant transmission through the window. The transmitted solar radiation gain is first absorbed by the enclosing surfaces and furnishings, then transferred to the air by convection and finally extracted by the cooling unit. Similarly, heat conducted to the inside surface of the exposed wall may be transferred to the cooling unit via the other enclosing surface by radiant and convective exchange. The actual heat gain to the room and the cooling unit load will then be influenced by the thermal coupling between the enclosing surfaces and the thermal capacity of the walls, floor and ceiling. Methods by which this influence may be assessed quantatively are urgently required. Work along these lines is strongly recommended.

HARRY BUCHBERG,* Los Angeles, Calif. (WRITTEN): Judging from the first sentence of the paper, the author is unaware that the network method of radiation analysis, as developed by Oppenheim (Reference 1), is primarily a method of considering radiation exchange between *all* isothermal surfaces making up an enclosure. The analysis is not confined to net transfer rates between pairs of surfaces as implied by the author. A radiation exchange network accounting for direct radiation exchange as well as all interreflections between any number of surfaces was developed and applied to a test room as presented in ASHAE TRANSACTIONS Vol. 61, 1955, p. 373, by the discusser. The author will note that the net radiation lost by the jth surface turns out to be $A_j\epsilon_j/\rho_j\,(W_j - R_j)$ where W is the black body emissive power (σT^4) and R is the radiosity; A, ϵ, and ρ are as defined in the paper being discussed. This relation for the rate of loss from the $_j$th surface is comparable to Equation 1. The real difficulty in applying such equations is the dependence of emissive power on the temperature of the surface raised to the 4th power. The temperature of the surface in turn depends upon the radiation exchange taking place and in addition upon all other heat flow paths through the construction and to the space by convection. Presentation of so-called absorption factors (a considerable job in itself) does not alleviate the main difficulty. In the previous ASHAE reference given, there is presented a method of coupling the radiation exchange network to the other heat flow paths making possible the accurate determination of radiant loss rates at each surface as well as many other useful quantities. Additional work should be done to determine the importance of including the radiation network in different situations and to determine when simplifications can be made.

For those who wish to refer to the ASHAE paper cited, attention is called to an error on p. 375 due to an inadvertent switch in the definition of emissive power, W. In this derivation W was initially defined as the black body emissive power (σT^4) in Equation C-1. This definition must be used consistently, therefore, ϵ must be struck out of the denominators in Equations C-13 and C-14. Also one ϵ should be struck out of the denominators in the values for the resistances shown on the diagram.

T. C. MIN, Auburn, Ala. (WRITTEN): The author is to be commended for his contri-

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bution introducing a new concept which leads to a method for calculating the rate of loss of a surface in an enclosure involving multi-reflection and multi-absorption.

Heating, refrigerating, and air-conditioning engineers particularly welcome the way this paper is presented. It is a modified version derived from the author's previous paper with more emphasis in the application to panel heating calculation. In his original paper, which was presented at the *ASME* meeting about a year ago, rigid mathematics is used freely. The example given in this paper should be most illuminating to the engineers in this field.

The author developed an ingenious quantity, namely absorption factor, defined as the fraction of the total emission of one surface which is absorbed by another surface including all paths whereby the energy may travel. He also defines the rate of loss of a surface as the rate of radiant energy emission by the surface minus the total rate of absorption of radiant energy at the surface. This rate of loss of a surface has, indeed, direct physical significance. It is also stated in the paper that the net transfer rate between two of the surfaces of an enclosure made up of many surfaces has no physical significance. It appears to the discusser that the net transfer between pairs of surfaces of an enclosure is meaningful and is just another useful tool in calculating the rate of loss of a surface in the enclosure.

The net transfer between two surfaces may be defined as the difference of the relative radiocities as used in the radiation network method†, *i.e.*,

$$q_{1-j} = (J_1 - J_j)A_1F_{1j} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (A)$$

where

q_{1-j} = the net radiation transfer between surfaces 1 and j.

J_1, J_j = the radiocity of surface 1, j.

F_{1j} = the angle factor of surface 1 and j.

It may also be defined as the difference of the total amount of radiation omitted by one surface and absorbed by the other surface, *i.e.*,

$$q_{1-j} = q_{1j} - q_{j1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (B)$$

where

q_{1-j} = the total amount of radiation emitted by surface 1 and absorbed by surface j per unit time.

This is used in a numerical method by Chao* and Sartain‡.

Applying the Kirchhoff's Law in the network diagram†, it is seen that the rate of loss of a surface, say j, of the enclosure is equal to the sum of the net transfer rates between surface j and surface 1, surface j and 2, j and 3, and so forth. Likewise,in Professor Chao's method, by the principle of conservation of energy, the same is true, *i.e.*

$$q_j = q_{j-1} + q_{j-2} + q_{j-3} + \cdots\cdots + q_{j-n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (C)$$

where

q_j = the rate of loss of surface j.

By substituting the Equation B which defines the net transfer between two surfaces to Equation C which is the expression for the rate of loss of a surface in an enclosure, it can be shown that

$$\begin{aligned} q_j &= (q_{j1} - q_{1j}) + (q_{j2} - q_{2j}) + \cdots\cdots + (q_{jn} - q_{nj}) \\ &= (q_{j1} + q_{j2} + \cdots\cdots + q_{jn}) - q_{1j} - q_{2j} - \cdots\cdots - q_{nj} \\ &= W_jA_j - \sum_{i=1}^{n} q_{1j} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (D) \end{aligned}$$

† A. K. Oppenheim: Radiation analysis by the network method (*ASME Transactions*, Vol. 78, 1956, p. 725).

* B. T. Chao: *On a Generalized Procedure for the Calculation of Thermal Radiation Transfer* (American Society of Mechanical Engineers, *Paper* No. 55–SA–61, 1955).

‡ E. L. Sartain: *Generalized Interchange Factor for Calculating Radiant Heat Transfer* (Unpublished thesis, University of Illinois, 1954).

and Equation D is identical to the Equation 1 in the paper. In other words, this indicates the net transfer rate between pair of surfaces is the component of the rate of loss of a surface of the enclosure.

In the other papers*,‡, employing the net transfer between two surfaces and the reflection factor, the major inconvenience involves the calculation of reflection factors. While in the author's paper, the major work is in connection with the calculation of absorption factors. It may be added that the concept of net transfer between two surfaces can be applied in radiation network and may also be used as a measure of the effect to the rate of loss of a surface in the enclosure.

These comments, are not intended to detract from the merit of the author's work, rather to point out the relation between the net transfer between two surfaces and the rate of loss of a surface in the enclosure, or between 2 parallel methods of approach.

There are two minor points about which I would like to question the author:

1. In the Table 1, why is the angle factor for outside wall to floor not the same as the angle factor for outside wall to ceiling?
2. Could the reflectivities and emissivities of the adiabatic surfaces 4, 5, 6, 7 in Table 1 be 1 and 0 respectively instead of 0.1 and 0.9?

Again congratulations to the author for introducing a valuable concept of absorption factor and an extension of the reciprocity theorem to non-black surfaces.

AUTHOR'S CLOSURE: The author expresses his appreciation to the discussers for their interest. He also thanks them for helping to relate this method of analysis to existing ones and for stressing the other problems of calculation such as those which arise due to the simultaneous transfer of heat by several modes.

Mr. Solvason emphasizes the complexity of the real problem and the author agrees with his point that further study is necessary. The major problem remaining is the linearization of the combined mode transfer equations. It is believed that the use of the method demonstrated in this paper will facilitate this step because of the basically simple nature of the radiation quantities defined. Since this paper did not propose to treat this problem, the case chosen for analysis was arranged to demonstrate the radiation method. The effects of other modes were not included.

Two comments can be made on Professor Buchberg's discussion. *First*, a number of methods exist for calculating radiant exchanges in gray surface enclosures and any method, in order to be correct, must take into account *all* surfaces. This is granted by all. My comment referred to where the emphasis is placed and was intended to contrast the points of view. *Second*, as stated, the fourth power in the radiation equations is a difficulty, as are also the non-integral powers in some convection equations. It is not likely that any method will come into everyday use for air-conditioning calculations until some simplifications are made.

The developments presented in Professor Min's discussion, which relate the methods of Chao and Oppenheim to this one, are valuable additions to the paper. This will be helpful to those who wish to compare the different points of view. It is a good place, incidentally, to compare the complexity of using the n^2 heat exchange quantities as is done in existing methods with the simplicity which results from combining all heat transfer information into n quantities. The author can see no justification, either from the theoretical or practical point of view, for using n^2 quantities to express the information which may be given by n quantities.

In answer to the specific questions about numerical values, in Table 1 the angle factor from the outside wall to the floor is given as 0.308 and to the ceiling as 0.304. Professor Min is correct, these should be equal. They should both be 0.306. This difference occurred because the sum of each row must be 1.0. Since F_{14} was the last quantity calculated, the roundoff of other quantities in the table was evidently concentrated here.

The reflectivities and emissivities of the adiabatic surfaces 4, 5, 6 and 7 listed in Table 1 are those of actual surface materials. If one wishes to find the loss rates of the other surfaces, the use of 1.0 and 0 for the adiabatic surfaces obviates the necessity of calculating the temperatures of these surfaces from heat balances. If one wishes to calculate the temperatures of the adiabatic surfaces, any value of reflectivity may be used, other than 1.0, as shown by an unpublished analysis.

1668

No. 1668

THE MORNING AFTER FOR EDUCATION

By Dr. Eric A. Walker*, University Park, Pa.

IN *Tristram Shandy*, Laurence Sterne, the eccentric eighteenth-century English novelist and clergyman, wrote that *The ancient Goths of Germany . . . had a wise custom of debating everything of importance twice—once drunk and once sober; drunk that their councils might not lack vigor; sober that they might not lack discretion.*

A little over two years ago, education began to emerge as a topic of primary interest to the American people. Fanned by events of grave national importance, this interest soon resulted in a debate that has often been conducted more along the lines of an argument than along those of an examination or a discussion. Almost everyone has become involved in this debate. A vocal naval research officer made his name known throughout the land by advocating a return to the *three R's*. A nationally syndicated political analyst found virtue in the Little Rock situation because it forces the students into private schools, which are, he claims, superior to the public ones. Industrial leaders delivered speeches extolling the value of a liberal education while their recruiters continued to search for technically trained graduates. A famed economist has advanced a *buy-now, pay-later* plan to finance higher education. The list could be extended almost endlessly.

Of course, those charged with the responsibility for the education of our youth—the teachers and administrators of our schools, colleges, and universities—added their voices to the debate. Perhaps because of their many years of comparative isolation from public scrutiny, they seemed unprepared for this opportunity to explain their work and their needs and to enlist popular support for their programs. Rather than deal with fundamental issues, many of them chose instead to parade their quarrels with others of their profession and to air their private squabbles.

The debate has centered around such subjects as quality versus quantity, the sciences versus the humanities, the *three R's* versus progressive education, public support versus private. Undergirding everything else has been the question of who is to pay for education in our country—and this question, in turn, has im-

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An address delivered at the 65th Annual Banquet of the American Society of Heating and Air-Conditioning Engineers, Philadelphia, January 1959.

plications concerning the objectives of education. Even the previously accepted position that mass education is requisite to the proper operation of a democracy has been challenged. Overall, the debate has been marked more by prejudice than by knowledge, more by opinion than by fact, more by vested interest than by public concern.

It has certainly been vigorous, and it has not been entirely sober. But we can no longer afford to indulge ourselves. It's time we took the second look—the sober one—to find out what the debate has so far produced, to gather some hard facts about American education, to find out what we need to do to improve our educational offerings, and to set about doing these things calmly and energetically. We must find out where we are and where we wish to go—and then we must chart a course that will get us there. Primarily, we must come to some hard decisions about what we teach; how we teach it; to whom we teach it; how much we should support the teaching of it, both financially and morally; and how that financial support is to be channeled from the people of our country to our schools and colleges.

The debate started when people began to realize that the boys and girls that were forcing the elementary schools onto double shifts throughout our country would one day be ready to enter college. Statisticians, working from census figures, soon pointed out that our colleges and universities would have to enroll twice as many young men and women by 1970 as were enrolled in 1955 simply to maintain the same ratio of college going. Others were quick to point out that the ratio has been steadily increasing for several years; and, because of the increasing complexity of our civilization, we could expect it to continue to increase. Consequently, *more* than twice as many young Americans will be clamoring for college entrance in 1970 as were enrolled in 1955, they pointed out.

The implications of these computations were soon apparent to everyone who read a newspaper or a magazine, listened to a radio station, or watched television. They meant that we should have to build within just 15 years as many classrooms, laboratories, libraries, and residence halls as were erected in the entire 300-year history of American higher education up to that time. It meant that, if these youngsters were going to be taught through traditional methods and techniques, we should have to double the faculties and staffs of our colleges and universities. And the cost, it was quickly pointed out, would be more than twice the present annual cost of our programs of higher education. For one thing, the expansion would have to be carried out during a period of continuing inflation, with steadily rising costs along the line. For another, teachers had not shared proportionately in the general increase in purchasing power the American people had been experiencing since the conclusion of World War II, and wages and salaries for faculty and staff members would have to be increased significantly if we were going to be successful in encouraging young men and women to enter teaching and in retaining those who were already on our faculties. In fact, it very much looked as though the annual bill for higher education would jump from the $3 billion it cost us in 1955 to at least $6 billion and perhaps to as much as $10 billion by 1970.

The American people became alarmed. The President of the United States appointed a committee to study the situation. Most of the states started studies of their own. National magazines carried articles pointing out that, after all, not everyone *had* to go to college. Others pointed out that we were already desperately short of professional workers of all sorts—especially, perhaps, of engineers and scientists. Business and industry spokesmen warned that commercial in-

terests could not be expected to give more than about one percent of the five per cent allowed by the tax laws. State legislatures found that taxes were already high and that it would be difficult or impossible to raise them further. Parents pointed to the steadily increasing costs of college going and maintained that further increases would restrict college training to the sons and daughters of the wealthier people. The students themselves resisted the idea of borrowing to pay for their education, pointing out the difficulty of establishing a family and starting a career even without the added handicap of a large debt. Almost no one seemed to favor considering Federal aid.

Then, on October 3, 1957, Russia launched the first of her several Sputniks, and the debate took on an entirely new dimension. Only four months earlier, an American magazine had carried an article describing the *six* satellites that we planned to launch during the International Geophysical Year. The article had gone on to report that Russia had also announced plans to launch *a* satellite. The Soviet satellite, the article reported, *sounded remarkably like a description of the U. S. satellite, translated into metric units.* In fact, it differed from the ones we planned to launch in only one respect, the article said: it was about five times as heavy as ours. But the article made it clear that we were not particularly worried about this difference. *U. S. Scientists,* the article stated, *suspect that some Soviet publicist made the mistake of multiplying by 2.2 instead of dividing by 2.2 when he tried to convert pounds into kilograms* when copying the American design.

Launched against this background of smug complacency, Russia's demonstration of its technical competence—made before we were ready even to attempt a launching—came as a tremendous shock to the American people. Where had we failed? What had happened to the technical supremacy in which we had put so much faith? For many, the answer lay in deficiencies in American education.

In this way, the question of *quality* was introduced into the debate, which had been concerned largely with *quantity* up to this time. Many, pointing out that thousands of craftsmen could not make the significant scientific discoveries that could be made by one well-trained genius, advocated that we junk our concept of universal education in order that we might concentrate our energies and our money on the education of a small intellectual elite. American education was compared with education in Russia and in western Europe—and many thought that it came off second best. It was charged that our schools were teaching *social adjustment* to the exclusion of basic studies, that we were entertaining students rather than educating them. Others pointed out that our programs for exceptional children were lop-sided: although we have done much for the mentally retarded child, they said, we have done almost nothing to make it possible for us to harvest the potential of our exceptionally brilliant children. The small private schools and colleges soon began to associate quality with size in their brochures and annual reports, and the larger ones listed the maintenance of quality to explain their lack of plans for expansion in the years ahead.

Because the Russian threat seemed to be concentrated in technological and scientific areas, many advocated programs to enlarge and to improve training in the sciences and in engineering. The reaction was immediate, and the war between the sciences and the humanities was fanned into renewed vigor.

As I said, it has been a whale of a party. But there's work to be done. It's time that we took the second look at our problems so that we can get on with this work.

What, exactly, has the debate produced so far? Well, for one thing, it has produced the single most important piece of Federal legislation concerning education in the United States since Abraham Lincoln signed the Morrill Land-Grant Act into law 97 years ago. This was the National Defense Education Act of 1958. Without going into details, I should like to outline for you the main provisions of this Act in order that I might explain why I think it is so important.

Two of the nine *working* titles of the Act are aimed at strengthening instruction in the sciences, mathematics, and modern foreign languages in the elementary and secondary schools and in the junior colleges of our country. This is to be done, in part, through grants to state educational agencies and loans to private schools for purchasing and installing special equipment needed for instruction in these areas. In part, it is to be done through grants to colleges and universities to help them set up centers for teaching modern foreign languages and institutes for providing advanced training for modern-language teachers. To encourage participation in these programs, grants-in-aid are to be given to teachers returning for refresher training at the institutes and to students in the language centers who promise to enter *service of a public nature.*

Another title is aimed at strengthening the guidance, testing, and counseling programs in the secondary schools. This is to be done by providing grants to help state educational agencies set up and operate significant guidance and testing programs in the public schools and by letting contracts with institutions of higher education for the operation of institutes to train guidance and testing personnel. Trainees in such institutes will receive grants to help defray the cost.

Title II provides for Federal help in granting low-interest loans to full-time college students. A student who qualifies may borrow as much as $1000 during any one year of his college career or as much as $5000 during his total college program to help meet the cost of his studies. Significantly, the Act states that *special consideration* should be given, in granting the loans, to students who expect to become teachers or who show special aptitude for the sciences, mathematics, or modern foreign languages. Further, up to one-half of the loan may be cancelled if the holder of the loan joins the faculty of a public school upon his graduation.

Another title is designed to help state agencies finance programs to train *highly-skilled technicians in . . . occupations requiring scientific knowledge*—that is, programs aimed at preparing people for semi-professional positions. This is the type of training normally received at a technical institute—the type for which we have designed, at Penn State, a two-year program culminating in an associate degree.

Title IV is aimed at strengthening post-baccalaureate training in institutions of higher education. This is to be done by establishing 1500 National Defense Fellowships, each with a value ranging from $2000 the first year to $2400 the third year, with an additional grant of $400 a year for each dependent. In granting the fellowships, preference is to be given those candidates who are interested in joining the faculty of a college or university after the completion of their studies. Further, in order that the fellowships may result in increased facilities for graduate training of college teachers, they are to be granted only for new or expanded programs.

Title VII authorizes the Commissioner of Education to award grants and contracts for research and experimentation in more effective methods for using television, radio, motion pictures, and other such media for educational purposes.

Another title is aimed at improving the adequacy and reliability of educational statistics provided by state and local reports and records and at strengthening

methods of collecting, processing, and disseminating those statistics. This part of the National Defense Education Act will be carried out through grants to help state agencies with on-going or new programs in this area.

Finally, the Act authorizes the establishment of a science information service in the National Science Foundation. This title recognizes that much scientific information is lost or duplicated because of ineffective methods of collection and dissemination. The program to be set up will include indexing, abstracting, translating, and all other services leading to more effective distribution of scientific information.

The importance of this Act cannot be measured by the amount of money it may pump from the Federal treasury into state and local agencies, just as the importance of the Morrill Act cannot be measured by the size of the original Federal grants for establishing land-grant institutions. The Morrill Act was important because it fostered a new concept of higher education in America—a concept that made it possible to match the actual educational needs of the country with programs that could satisfy those needs. The National Defense Education Act is important for the same reason. It is important because of the direction it gives—because of the concept it fosters.

What is that concept? Let's look at the Act again—it recognizes that:

It is in the national interest to strengthen instruction in certain specific areas in the elementary and secondary schools.

We need to establish a systematic method of identifying our talented youth early enough to direct their energies toward their most useful social roles.

We must make it financially possible for capable young people to attend college.

We need a multi-level collegiate structure that will permit us to develop the intellectual potential of each boy and girl to its highest potential. Further, to achieve such a system, we must strengthen technical-institute training on the one hand and graduate, post-baccalaureate instruction on the other.

We must encourage young people to pursue careers in teaching, at both the school and the collegiate levels.

We need to conduct research to adapt new methods of mass communication to serve educational objectives.

We must have better information and more systematic data for solving the difficult problems that lie ahead of us.

These are sound objectives. And the program outlined in the Act makes a good beginning at establishing the machinery for realizing them. If adequately supported, philosophically as well as financially, it will go a long way toward solving some of our most pressing problems.

But what else has the debate accomplished? Not much, I'm afraid, except to generate heat and confusion. To convert this random energy into useful work, I want to suggest certain basic considerations for guiding our second—our sober—look at our problems.

In the first place I think we must not confuse the nature of our real challenge. It is vitally important for us to protect our ideals of democracy, of free enterprise, and of the dignity of man—for us to meet the Russian challenge. But we have a higher goal than that ahead of us—the creation of the best possible life for the greatest possible number of people. For the achievement of this higher goal, we must look at ourselves, not at the Russians, for our success must be measured in terms of achieving our own highest potential, not in terms of matching the Russian performance.

When I was in high school, I went out for track. I didn't set many records, but I learned an important lesson. I learned that I could not achieve my best performance by running my competitor's race. When I let the other fellow set the pace, my own time invariably suffered.

The goal we have before us is worthy of the best efforts of all of us. We are at the threshold of a new world that promises rewards that stagger the imagination. It has already given us the ability to change, rather than merely to adapt, nature to suit our needs. It has already given us a source of energy that will probably solve for all foreseeable time our power needs. It has already given us cures or preventions for some of man's most persistent and most deadly ills. It has already given us the ability—if only on a small scale—to explore the space seas lying about our island world. It has already given us automation, which promises at long last to free man from manual labor. It has already given us more material wealth and creature comforts than any other people have ever known on the face of the earth. And it has already shown us that the discoveries made so far are like the visible part of an iceberg—that which lies hidden beneath the surface bulks larger than that which is now known.

Our stake in this brilliant new age of science is measured by our ability to know and to understand—by our capacity for mental, intellectual, and moral development. To achieve it, we must run our own race—not that of our competitor.

Further, I think we must realize that we do not have a choice between quality and quantity. We must have as much of both as we can manage. We must have quality *in* quantity.

It is true that one well-trained genius can advance the frontiers of science farther than can a thousand craftsmen, but it is also true that two well-trained geniuses can cover twice as much ground. Then, too, as in mountain climbing, the farther the explorer advances, the more numerous become the helpers he must have and the more rigorous become the demands made upon those helpers. Yesterday, the scientist processed his own data and performed his own computations. Today, he must turn to giant electronic computers for this work. Yesterday, he needed only a microscope, a balance, a stop watch, and such equipment for his work. Today, he needs a nuclear reactor, a multistage rocket, an infrared spectrophotometer. Because of changes such as these, the scientist today must be supported by an army of engineers and technicians, all working at levels that make new and increasingly exacting demands on their skill, ingenuity, and training.

In fact, a central characteristic of the age in which we are living is that we replace, at the level of routine chores, human labor with machines that do the work faster, more accurately, and at less cost. And everyone, all along the line, must operate at a higher human level of mental activity. As each new discovery is reduced to practice and added to the general culture, all people must bring to the job of living a higher level of intellectual response. The ditch digger must learn to operate a complex power shovel. The teamster must accept the tremendous responsibility of operating 30 tons of steel and rubber propelled at high speeds by a 200-hp engine over crowded public highways. The voter, whatever his occupation, must be able to form intelligent opinions about Quemoy as well as about local tax issues, about Little Rock as well as about the qualifications of a candidate for the school board, about the Near East as well as about the advisability of employing a city manager.

Education, it is clear, must be provided at increasingly higher levels to increas-

ing numbers of people. Dr. John Gardner, president of the Carnegie Corporation of New York, has reminded us that we must have both good plumbers and good philosophers. If we don't, neither our pipes nor our ideas will hold water. We must have both quality *and* quantity. We must have quality *in* quantity.

Finally, I think we must realize that our schools, colleges, and universities do not operate in a social vacuum—that general attitudes held by the public—by you people, if you wish, and your neighbors—have much to do with the kind of job the schools, colleges, and universities can—and will—do for your children. Our pupils and students—at whatever level—simply are not generally working at the level of full intellectual capacity. All of you here this evening know that solving a difficult mathematical problem or discovering the perfect order of any interlocking subsystem of nature can be as exciting as scoring a winning touchdown or having your first date. But our children don't seem to know it.

We cannot expect our children to make any particularly significant sacrifices for, or to be strongly motivated toward, a life of scholarly activity when, as a society, we honor our professional football players and our ballad singers more than we honor our scientists and our scholars. In my travels about the country, I am often pulled aside at my meetings so that someone can tell me about a particularly promising football prospect. I have yet to be pulled aside to be told about a particularly promising scholar. I suspect that, because of the help of volunteer recruiters, almost no potential football talent is allowed to blush unseen and unscouted. I wish I could suspect the same thing about potential talent for scholarship.

We cannot afford to support this anti-intellectualism. It stands between the student and the intrinsic appeal of science and scholarship; the feeling of awe before their universalized observations; and the satisfying *rightness* of their workmanship, their order, and their organization. It stands between our youth and their full development as mature, productive members of society.

It also stands between our educational institutions and an adequate program of support for them. We must remember that the support for our schools and colleges must come from the people. There simply is no other source for this support. And it is impossible for me to believe that a people who can afford to bet $2¼ billion a year at the horse races and spend $10½ billion a year for liquor cannot afford to finance the best possible education for all young men and women who can profit from it. We can afford it if we want to afford it.

To the extent to which the Russian challenge has dissipated our lethargy, it has been good for us. To the extent to which the problem of numbers has focused attention on needed reforms in our educational offerings, it, too has been good for us. Perhaps even in the triggering of the Great Debate, both these things have been good for us. The debate has been carried out vigorously, if not entirely soberly. It has even resulted in an extremely important piece of Federal legislation.

But the time has come, it seems to me, for us to add sobriety and clear thinking to our deliberations. We are in danger, I believe, of proving the old saying that he who sins when drunk will be punished when sober. It is time for us to concentrate our energies on solving our basic problems, on charting our future course, on making basic decisions.

But the debate has not been fruitless. The pessimist would say that we've covered only half the distance we've got to go. I prefer, with the optimist, to say that we're already half way to our destination. In the second half of our journey, we must keep, I think, at least four things before us as guides:

1. In our deliberations, we must set our sights higher than mere survival. We must aim at realizing the full potential of our own genius in order that we may reap the full harvest of the scientific age for the benefit of all mankind.

2. We must realize that we cannot achieve this goal by concentrating our educational energies on the production of an intellectual elite. The challenge demands no less than the best possible education for all people in terms of the capacity and ability of each person.

3. We cannot expect our educational institutions to accomplish this goal in a social climate of anti-intellectualism. If, as a society, we wish to increase the vigor of our educational programs, we shall have to honor scholarship.

4. And we cannot expect to achieve this goal unless we are willing to channel a greater proportion of our wealth into our educational institutions.

1669

No. 1669

ASHRAE ANNUAL MEETING, 1959

LAKE PLACID, NEW YORK

THE ANNUAL MEETING held at the Lake Placid Club, June 22 to June 24, was the first Meeting to take place following the consolidation of the two constituent societies into the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS. The program for the meeting (see page 343) was a crowded one since it was arranged to accommodate a business session, a total of 8 technical sessions, 6 forum type sessions and 3 conferences, along with many meetings of committees. Concurrent sessions were scheduled in order to fit the sessions into the time available and there were 2 technical sessions and a conference on each of the 3 mornings during the meeting. Afternoons were devoted largely to committee meetings, to the annual golf tournaments, and to a full program of special events including trips for the visiting ladies and children.

With so large a number of sessions on the program, the number of technical papers was also large since a total of 29 technical papers were programmed and presented, and are included in this volume of TRANSACTIONS. Additionally, the program included a total of 11 papers prepared for and presented at the 3 conference sessions.

Attendance figures showed that a total of 769 were registered, and which included 426 members, 89 guests, 254 ladies and children.

At the opening of the Annual Meeting at 9:00 a.m. on Monday, June 22, President Cecil Boling, West Hartford, Conn., presented R. A. Baker, Syracuse, N. Y., chairman of the Meeting Arrangements Committee, who outlined the meeting plans and expressed the pleasure of the host chapter in having the meeting held at Lake Placid. President Boling then outlined some of the activities of the Executive Committee and explained that that committee had found it necessary to take many interim actions which have met with the approval of the Board of Directors. He expressed his appreciation of the fine co-operation he had received during an especially difficult period. He then introduced First Treasurer J. H. Fox, Toronto, Ont., Canada, who outlined the financial situation of the Society. President Boling directed attention to the United Engineering Center project scheduled for completion in 1961, and a colored slide of the planned structure was shown. The business session was then recessed until the time of the Welcome Luncheon.

At the Welcome Luncheon held on Monday at noon, President Boling presided and extended the official welcome to all present. He then introduced the members of the Meeting Arrangements Committee. Chairman S. J. Williams, Cleveland,

Ohio, of the Program Committee then presented the various authors and conference speakers. President Boling briefly reviewed the principal events of the preceding 6-month period. The new officers as listed in the agreement for consolidation for the period from June 1959 to January 1960 were installed in office.

The Banquet was held on Tuesday evening with the newly installed President A. J. Hess, Los Angeles, Calif., presiding. Featured at the banquet was the presentation of the presidential gift to retiring President Boling in accordance with the established custom of ASRE. Two existing ASRE awards were also made with the Wolverine Award for 1958 going to Professor H. M. Hendrickson, Seattle, Wash., while the Klixon Award for 1958 was presented to H. O. Spauschus and R. S. Olsen, Louisville, Ky.

As will be noted from the program, many committee meetings were held during the course of the Annual Meeting. Included were the Finance, Executive, Research and Technical, Exposition, Regions Central, Charter and By-Laws, Program, and Publications Committees. The Board of Directors held two meetings.

The first technical session, held in the Tea Room at the Club, was opened by President Boling at 9:30 a.m. on Monday morning, and was a Topical Session on Compressors. With S. J. Williams acting as moderator, the 3 papers programmed for the session were presented and discussed, and the session was adjourned at 12:30.

First Vice Pres. Hess called the second technical session to order at 9:30 a.m. on Monday in the Agora Auditorium at the Lake Placid Club. The session was organized as a Topical Session on Ventilation and G. B. Priester, Baltimore, Md., served as moderator. Following the presentation and discussion of the 4 papers on the session program, Mr. Priester adjourned the session.

The Domestic Refrigerator Engineering Conference with F. A. Noll, Cedar Rapids, Ia., as chairman was held on Monday, June 22, at 9:30 a.m. in the Olympic Arena in the village of Lake Placid. Lt. Col. G. E. Danald, Associate Director, Food Radiation and Preservation Division, Quartermaster Food and Container Institute, Chicago, Ill., delivered the keynote address, using as his subject the preservation of food by radiation. The topic of the conference was "Should the Icebox Return?—The basic designs and functions required for the present day refrigerator," and 3 papers on this topic were presented and discussed. The papers are not included in the TRANSACTIONS, nor are the discussions which took place on the floor of the conference. No bulletin has been prepared containing them. Resumés can be found on page 65, ASHRAE JOURNAL, June, 1959.

During the afternoon of Monday, June 22, a series of 6 Forum type sessions took place under the general direction of E. N. Johnson, Chicago, Ill. No record of the proceedings at these Forums is kept.

The third technical session on Tuesday, June 23, at 9:00 a.m. was in charge of First Vice Pres. D. D. Wile, Los Angeles, Calif., and was organized as a Topical Session on Noise, with W. E. Fontaine, W. Lafayette, Ind., as moderator. Four papers were presented; 3 of them were discussed.

After opening the fourth technical session at 9:00 a.m. on Tuesday, June 23, Second Vice Pres. Walter A. Grant, Syracuse, N. Y., presented R. A. Line, St. Louis, Mo., who acted as moderator of the session. Three papers had been prepared for this Topical Session on Heat Transfer and all were presented and discussed.

The Industrial Ventilation Conference, held on Tuesday, June 23, at 9:00 a.m., was presided over by G. B. Priester, Baltimore, Md. Four papers had been

prepared for this conference and were presented and discussed. These papers together with a digest of the discussion which took place are included in a bulletin available from the Society. None of these papers are printed in TRANSACTIONS. Resumés can be found in the ASHRAE JOURNAL, June, 1959, on pages 66 and 67.

The fifth technical session was opened by Third Vice Pres. R. H. Tull, Springfield, Mass., at 1:30 p.m. on Tuesday, June 23rd. This was a general session and R. S. Buchanan, Columbus, Ohio, presided as chairman. The 3 programmed papers were presented and discussed.

Following the opening of the sixth session on Tuesday, June 23 at 1:30 p.m., Fourth Vice Pres. John Everetts, Jr., Philadelphia, Pa., called on H. P. Harle, Louisville, Ky., to take charge as moderator. The session was organized as a Topical Session on Sound Level and 3 papers were presented and discussed.

With Second Treasurer J. H. Fox in charge, the seventh technical session opened at 9:00 a.m. on Wednesday, June 24, and he turned the conduct of the session over to J. W. May, Louisville, Ky., who was the moderator of this Topical Session on General Refrigeration. The 4 papers prepared on this topic were presented and discussed.

The eighth technical session was organized as a Topical Session on Air Conditioning, and was opened by First Treasurer, F. Y. Carter, Detroit, Mich., at 9:00 a.m. on Wednesday, June 24. W. L. McGrath, Syracuse, N. Y., was presented to act as moderator and 4 papers were presented and discussed.

The Cryogenics Conference, which was opened at 9:00 a.m. on Wednesday, June 24, had as its topic: "Closed Circuit Transportation of Cryogenic Fluids." V. J. Johnson, Denver, Colo., was the chairman and 4 papers prepared for this conference were presented and discussed. In accordance with usual Society practice, these papers are not included in the TRANSACTIONS. The papers and a digest of the discussions which took place are available in a separately printed Bulletin. Resumés of the papers appear in the ASHRAE JOURNAL, June, 1959, pp. 68 and 69.

PROGRAM—ANNUAL MEETING

Lake Placid Club, Lake Placid, N. Y.

June 22–24, 1959

Saturday, June 20

9:00 a.m.	Finance Committee
12:00 noon	Executive Committee luncheon-meeting
1:30 p.m.	Research and Technical Executive Committee
3:00 p.m.	Exposition Committee
4:30 p.m.	Regions Central Committee dinner-meeting

Sunday, June 21

10:00 a.m.	REGISTRATION, Main Floor Lobby
	Research and Technical Committee
2:00 p.m.	Long Range Planning Committee
	Membership Development Committee
	Standards Committee

Sunday, June 21 (*Continued*)

2:30 p.m. Meetings Arrangements Committee
4:00 p.m. Board of Directors dinner-meeting
4:00 p.m. Reception
8:30 p.m. Adirondack Antics

Monday, June 22

7:30 a.m. Domestic Refrigerator Conference breakfast
Forum Moderators' breakfast
Speakers' breakfast—First Session
Speakers' breakfast—Second Session
8:00 a.m. Professional Development Committee breakfast
Public Relations Committee breakfast
8:30 a.m. REGISTRATION, Main Floor Lobby
Open "His" and "Her" Golf Tournaments
9:00 a.m. Annual Meeting of the Society, Agora Auditorium
Call to Order—Pres. Cecil Boling
Remarks by President Cecil Boling
Address of Welcome by R. A. Baker, Director, Region I
9:30 a.m. Tech. Comm. 1.7
9:30 a.m. FIRST TECHNICAL SESSION
Call to Order—Pres. Cecil Boling

TOPICAL SESSION ON COMPRESSORS

S. J. Williams, *Moderator*

An Analysis of Turbine Driven Centrifugal Refrigeration Systems, by W. G. Dorsey, Jr., Kansas City, Mo., presented by Mr. Dorsey

Effects of Refrigerant Properties on Centrifugal Compressor Impeller Dimensions and Stage Performance, by F. J. Wiesner, Jr., and H. E. Caswell, Syracuse, N. Y., presented by Mr. Wiesner

Free Piston Compressor in an Air-Conditioning System, by R. J. McCrory, R. W. King, and J. H. McNinch, Columbus, O., presented by Mr. McCrory

9:30 a.m. SECOND TECHNICAL SESSION
Call to Order—1st Vice Pres. A. J. Hess

TOPICAL SESSION ON VENTILATION

G. B. Priester, *Moderator*

Automatic Computer for Fan Testing, by C. H. Pountney, Jr. and D. W. Skipworth, Cleveland, O., presented by Mr. Skipworth

Evaluation of Filters for Removing Irritants from Polluted Air, by N. A. Richardson and W. C. Middleton, Davis, Calif., presented by Mr. Richardson

Fundamental Analysis of Chimney Performance, by W. G. Colborne, Windsor, Ontario, Canada, and W. C. Moffatt, Kingston, Ontario, Canada, presented by Mr. Colborne

A New Method of Chimney Design and Performance Evaluation, by W. C. Moffatt, Kingston, Ontario, Canada, and W. G. Colborne, Windsor, Ontario, Canada, presented by Mr. Moffatt

Monday, June 22 (*Continued*)

9:30 a.m. Domestic Refrigerator Engineering Conference
Call to Order—F. A. Noll, *Conference Chairman*
Conference Topic—Should the Ice Box Return?—The basic designs and functions required for the present day refrigerator.
Keynoter: Lt. Col. G. E. Danald
Control of Temperature, by E. A. von Arb
Humidity and Temperature Conditions, by F. P. Speicher and E. W. Zearfoss
Air Circulation in Freezer Compartment—Benefits and Problems, by C. H. Wurtz

12:45 p.m. Welcome Luncheon
Installation of Officers
Ladies and Children's luncheon, Marcy Hotel

2:00 p.m. Trip to Whiteface Mountain with side trips
Card Party

2:00 p.m. Forums—E. N. Johnson, *in charge*
1. Freeze Protection of Heating Coils
Moderator: Earl Williams
2. Immersion Chilling of Poultry
Moderator: E. M. Kerrigan
3. Research, the Engineer and the Appliance Industry
Moderator: E. A. Baillif

3:30 p.m.
4. Sound and Vibration Problems in Air Delivery Equipment
Moderator: J. B. Graham
5. Ice Maker Design Considerations
Moderator: D. E. MacLeod
6. Status of New Utilization Voltage Proposals
Moderator: A. S. Anderson

2:30 p.m. Charter and By-laws Committee
Education Committee
Emblem and Insignia Committee
Honors and Awards Committee
Program Committee
Publications Committee
Research Advisory Committees
Refrigeration
Sorption
Thermal Circuits

3:00 p.m. Technical Committee 1.5

4:00 p.m. International Affairs Committee

6:30 p.m. Outdoor Steak Fry—Followed by special program

Tuesday, June 23

7:30 a.m. Speakers' breakfast—Third and Fourth Sessions
Speakers' breakfast—Fifth and Sixth Sessions
Industrial Ventilation Conference breakfast
Tech. Committee 4.2 breakfast

8:30 a.m. REGISTRATION, Main Floor Lobby

Open "His" and "Her" Golf Tournaments

9:00 a.m. THIRD TECHNICAL SESSION

Call to Order—1st Vice Pres. D. D. Wile

TOPICAL SESSION ON NOISE

W. E. Fontaine, *Moderator*

Investigation and Control of Refrigerator Noise, by E. A. Baillif and J. P. Laughlin, Evansville, Ind., presented by Mr. Laughlin

Some Methods for Investigating Noise from Compressors Used on Household Refrigerators, by R. C. Binder, W. Lafayette, Ind., presented by J. B. Chaddock

Acoustical Testing of Mufflers for Halogenated Hydrocarbon and Oil Refrigeration Systems, by A. F. Martz, Jr., J. L. Martin, and W. R. Danielson, St. Joseph, Mich., presented by Mr. Martz

Suppression of Oscillations in Gas-fired Residential Heating Equipment, by C. F. Speich and A. A. Putnam, Columbus, O. (By title only)

9:00 a.m. FOURTH TECHNICAL SESSION

Call to Order—2nd Vice Pres. Walter A. Grant

TOPICAL SESSION ON HEAT TRANSFER

R. A. Line, *Moderator*

Solar Radiant Gains through Directional Glass Exposure, by R. D. Cramer and L. W. Neubauer, Davis, Calif., presented by Mr. Cramer

Load Calculations Using Pretabulated Admittance Functions, by W. B. Drake, Sunnyvale, Calif., Harry Buchberg, and D. Lebell, Los Angeles, Calif., presented by Mr. Buchberg

Transfer Admittance Functions for Typical Composite Wall Sections, by W. B. Drake, Sunnyvale, Calif., Harry Buchberg, and D. Lebell, Los Angeles, Calif., presented by Mr. Drake

Large-scale Wall Heat-flow Measuring Apparatus, by K. R. Solvason, Ottawa, Ontario, Canada, presented by Mr. Solvason

9:00 a.m. INDUSTRIAL VENTILATION CONFERENCE

Call to Order—G. B. Priester, *Moderator*

Design and Performance of Industrial Supply Air Systems, by J. H. Clarke
Dust and Fume Control Systems, by P. J. Marschall
General Ventilation in the Steel Industry, by F. E. Tucker
Radiant Heat Control in Industrial Plants, by W. G. Hazard

9:00 a.m. Technical Coordinating Committee

9:30 a.m. 24-57R Liquid Coolers Committee

COOPERATING COMMITTEES

AIEE—Hermetic Motor Insulation Materials (all day)
AIEE—Hermetic Motor Insulation Systems (all day)

10:00 a.m. Putting Contest for Ladies

12:30 p.m. Ladies Luncheon

Tuesday, June 23 (*Continued*)

1:30 p.m. Fifth Technical Session

Call to Order—3rd Vice Pres. R. H. Tull

General Technical Session

R. S. Buchanan, *Moderator*

Condensation Between the Panes of Double Windows, by A. Grant Wilson and E. S. Nowak, Ottawa, Ontario, Canada, presented by Mr. Wilson

Evaporative Cooling for Common Storages of Fruits and Vegetables, by R. S. Ash, Phoenix, Ariz., presented by Mr. Ash

Controlled Atmosphere Apple Storage Process and its Requirements on Refrigeration Structures and Systems, by J. W. Zahradnik, Amherst, Mass., presented by Mr. Zahradnik

1:30 p.m. Sixth Technical Session

Call to Order—4th Vice Pres. John Everetts, Jr.

Topical Session on Sound Level

H. P. Harle, *Moderator*

Proposed ASHRAE Standards for the Measurement of Sound from Equipment, by C. M. Ashley, Syracuse, N. Y., presented by Mr. Ashley

Experimental Study of Grille Noise Characteristics, by B. H. Marvet, Knoxville, Tenn., presented by Mr. Marvet

Effect of Fuel Composition on Oil Burner Noise, by W. A. Beach, R. W. Sage, and H. F. Schroeder, Linden, N. J., presented by Mr. Schroeder

2:00 p.m. RAC on Industrial Environment

Divisional Advisory Committee

Advertising Committee

2:30 p.m. Lake Placid Boat Trip for ladies and children

6:00 p.m. Cocktail Party

7:00 p.m. Banquet

Awards and Presentations

9:00 p.m. Dancing

Golf prizes

Wednesday, June 24

7:30 a.m. Speakers' Breakfast—Seventh and Eighth Sessions

Cryogenics Conference breakfast

8:30 a.m. Registration, Main Floor Lobby

9:00 a.m. Seventh Technical Session

Call to Order—J. H. Fox, 2nd Treas.

Topical Session on General Refrigeration

J. W. May, *Moderator*

Comparison of Fluoroalkane Absorption Refrigerants, by B. J. Eiseman, Jr., Wilmington, Del., presented by Mr. Eiseman

Estimating Water Content of Certain Dried but Uncharged Hermetic Refrigerating Compressors, by H. M. Elsey, J. B. Kelley, and R. B. Sharpe, Columbus, Ohio, presented by Mr. Elsey

Wednesday, June 24 (*Continued*)

Effect of Pressure and Temperature on the Viscosity of Refrigerants in the Vapor Phase, by O. W. Witzell and C. Z. Kamien, W. Lafayette, Ind., presented by Mr. Witzell

Repeated Scrape Abrasion Testing of Enameled Wires in Gaseous and Liquid Refrigerants, by F. F. Trunzo, G. J. Bich, and G. W. Hewitt, Pittsburgh, Penna., presented by Mr. Trunzo

9:00 a.m. EIGHTH TECHNICAL SESSION

Call to Order—F. Y. Carter, 1st Treas.

TOPICAL SESSION ON AIR CONDITIONING

W. L. McGrath, *Moderator*

Determination of the Effectiveness of Window Shading Materials on the Reduction of Solar Radiation Heat Gain, by R. C. Jordan and J. L. Threlkeld, Minneapolis, Minn., presented by Mr. Threlkeld

Heat Flow through Glass with Roller Shades, by Necati Ozisik and L. F. Schutrum, Cleveland, Ohio, presented by Mr. Ozisik

Evaluation of Three Room Air Distribution Systems for Summer Cooling, by J. J. Reinmann, Alfred Koestel, and G. L. Tuve, Cleveland, Ohio, presented by Mr. Reinmann

Surface Odor Adsorption and Retention Properties of Surfaces, by W. F. Hopper, Cleveland, O., presented by C. M. Humphreys

9:00 a.m. CRYOGENICS CONFERENCE

Call to Order—V. J. Johnson, Conference Chairman

CLOSED CIRCUIT TRANSPORTATION OF CRYOGENIC FLUIDS

Transfer of Cryogenic Fluids—Design Aspects of Transfer Lines, Valves, Couplings and Associated Components, by R. B. Jacobs

Cryogenic Pump Design, by L. P. Schafer, Jr.

Cryogenic Insulation, by R. H. Kropschot

Instrumentation Methods and Techniques for Monitoring and Controlling Closed Circuit Transportation of Cryogenic Fluids, by Clarence Gettelman

12:30 p.m. Executive Committee luncheon-meeting

1:00 p.m. Research and Technical Executive Committee

1:30 p.m. RAC on Heat Pump

RAC on Insulation

RAC on Heat Flow through Fenestration

2:00 p.m. Board of Directors

MEETING ARRANGEMENTS COMMITTEE

Honorary Chairmen

A. E. Stacey, Jr. C. M. Ashley Walter A. Grant

R. A. Baker, *General Chairman*
L. L. Schneider, *Entertainment*
J. R. Schreiner, *Reception*
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K. M. Gerteis, *Finance*
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Fred Hack, *Sessions*
A. L. Jones, *Transportation*
D. C. Raasch, *Banquet*
M. E. Barnard, *Ladies*

1670

No. 1670

AN ANALYSIS OF TURBINE DRIVEN CENTRIFUGAL REFRIGERATION SYSTEMS

By W. G. Dorsey, Jr.*, Kansas City, Mo.

MANY FACTORS enter into the analysis and determination of the type of refrigeration system to be used when steam is available from either existing or contemplated boiler capacity as a power source. This article points to the advantages of using steam turbine-driven centrifugals as compared with steam operated absorption refrigeration systems.

The factors to be considered are as follows:

- Operating Economy
  - Heat Balance
  - Installed auxiliary horse power
  - Pumping cost
  - Cooling tower fan horse power
  - Auxiliary services cost
  - Cost of operators
  - Space requirements
  - Ease of operation
- Maintenance Cost
- Noise and Vibration
- Installation Cost
  - Labor and supervision
  - Provisions in building for support
  - Controls
- First Cost
  - Amortization of first cost related to operating cost

All of these factors intertwine to evolve the overall cost of the system. In general, where steam pressures above 35–40 psig are available the turbine will prove to be the best application. These turbines are generally operated condensing at 26–27 in. Hg vacuum. At 125 psig inlet condition, steam rates of 12 lb per hp-hr on large horsepowers and 16–17 lb per hp-hr on small horsepowers are regularly available. Steam surface condensers for this steam consumption are available in sizes to match the turbine selection. Auxiliary horsepower for all sizes of turbine-driven centrifugals consists of ⅓-hp oil pumps and ½-hp purge unit. The oil pump will have a service factor of 1 whereas the purge compressor will operate about 20 percent of the time. Steam absorption machines, for example, require from 5-hp on the smallest size to 14½-hp on the largest size which operate continu-

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ously. The nominal steam rate for absorption machines using 12 psig steam is 19.4 lb per ton hr based on 45 F chilled water.

Turbine-driven centrifugals and the steam condenser require 3 gpm per ton of 85 F cooling tower water returning to the tower at approximately 105 F; whereas the absorption machines will require an average of 3.5 to 4 gpm per ton of 85 F water returning to the tower at approximately 101 F. The turbine-driven machine will generally require a smaller tower with less fan horsepower due to the lower quantity of water at a higher temperature head on the tower. Resultant windage and evaporation losses are also lower. Pumping costs for condenser water are lower as a result of the reduced quantity of water handled.

Since refrigeration systems do not operate at full load 100 percent of the time the part load performance becomes an important consideration in operation cost. Seasonal operation on air-conditioning applications will vary from 2000 hr per season in the warmer climates to 1200 hr per season in the northern areas of the U.S. It is rather difficult to pinpoint exact hours of operation at any given load point but in general it can be assumed that full load operation will occur only 20 percent of the time, 75 percent load for 30 percent of the time, 50 percent load—40 percent, and 25 percent load—10 percent.

Turbine-driven centrifugals at part load performance generally are highly efficient in that the horsepower required at part load falls off at a faster rate than does the tonnage, resulting in a low total steam consumption at part loads. In practice a further contribution to low steam consumption results from the fact that on those days that reduced loads are encountered the cooling tower water is below design temperature, which causes the steam condenser to operate at increased vacuum thus increasing the efficiency of the turbine.

Absorption machines on the other hand have a steam rate based upon ton-hr rather than hp-hr which results in the tonnage falling off at a rate greater than the total steam consumption. For example, at 75 percent tonnage the steam consumption would be 80 percent of full load consumption. (See Fig. 1.) Absorption machines cannot take advantage of reduced condenser water temperatures as by their nature they require a relatively constant temperature in the absorber and condenser and a throttling by-pass valve is always furnished with the machine to maintain this constant temperature. It should be noted here that absorption machines cannot operate at steam pressures higher than about 12 psig, consequently where steam pressures are higher in the distribution system a pressure-reducing station must be used which in itself is a loss of power in the heat balance. While there is some credit due to the absorption machine because of the fact that condensate is returned to the boiler at approximately 200 F while the steam condenser returns condensate at 120 F this credit is offset by the lower total steam consumption operating from distribution steam pressure on the turbine. The absorption machine must be charged with the cost to attain the distribution steam pressure and the loss of potential power through the pressure-reducing valve. Turbine-driven centrifugals less steam condenser require space as follows:

Smallest machines—8 ft wide x 14 ft-10 in. long x 6 ft-9 in. high
700 Ton—12½ ft wide x 19 ft-2 in. long x 9 ft-9 in. high

Steam condensers will require 2–3 ft width and about 12 ft length over the entire range. An allowance of about 10 ft at one end must be available for tube removal. Absorption machines for comparable tonnage to centrifugals:

Smallest machines—5 ft wide x 19 ft long x 8 ft high
Largest machines—6 ft-8 in. wide x 25 ft-6 in. long x 11 ft-3 in. high

An allowance of 16–20 ft at one end must be available for tube removal. When considering the total space required including tube removal allowance the total sq ft of floor space will be about the same with the absorption machine requiring a long narrow space and the turbine-driven machine space requirements being shorter in length and somewhat wider.

Turbine-driven centrifugals with their steam condensers are about 25 percent higher in operating weight than the absorption machines; however, this does not affect the overall cost except in the most critical of existing buildings for upper floor installation.

The question as to the type of operating personnel required often arises as well as whether any operator is required. Often, local codes require operators for refrig-

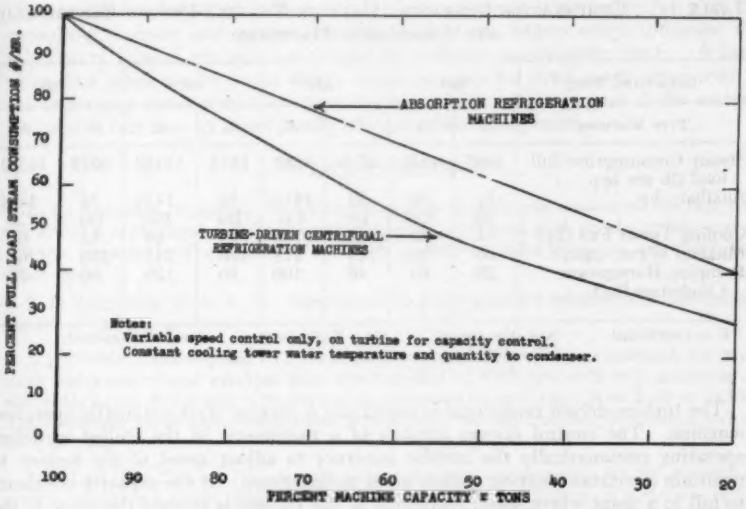

FIG. 1. . . . TYPICAL STEAM CONSUMPTION CURVES FOR TURBINE-DRIVEN CENTRIFUGAL AND ABSORPTION MACHINES

eration machines but exclude absorption machines by a technicality where water and/or lithium bromide is not included in the list of refrigerants. Reasonable people know, however, that for an investment of the sizes being discussed some provision is always made for operators—whether by building or plant maintenance people or by specific machine room personnel. These people are all that is necessary for operation and preventive maintenance on either centrifugals or absorption machines.

In most areas owners themselves recognize the need for someone to check the machines occasionally and in other areas union agreements call for some type of operating engineer regardless of type of machine.

The subject of noise and vibration is an involved subject and outside the scope of this paper, however, some generalities may be stated. Today's high caliber rotative machinery is both statically and dynamically balanced virtually eliminating

vibration. In addition, simple cork pads are generally installed to isolate the equipment from the building structure. As a result of this vibration has ceased to be any problem.

Noise level comparisons between centrifugals and absorption machines are misleading. Windage noises exist in turbines and compressors as well as gas noise in condenser and cooler. In the absorption machine steam condensate flashing and evaporation noises in the evaporator exist, so the comparison is really one of type and frequency of noise rather than the existence or nonexistence of noise. In nearly all cases either machine is installed in a machine room with other mechanical equipment such as fans and pumps, and elevator motors, so that the noise from the refrigeration machines becomes unimportant in the overall picture.

TABLE 1. . . . COMPARATIVE OPERATING DATA ON TURBINE-DRIVEN CENTRIFUGAL AND ABSORPTION MACHINES

CAPACITIES, TONS	200		430		600		700	
TYPE MACHINE	C	A	C	A	C	A	C	A
Steam Consumption full load (lb per hr).	3460	4230	6720	8880	8578	12450	9035	14300
Auxiliary hp	5/6 Int	10 Co	5/6 Int	14½ Co	5/6 Int	14½ Co	5/6 Int	14½ Co
Cooling Tower Fan (hp)	15	20	30	37½	45	60	45	60
Makeup Water (gpm)	66	90	115	175	180	245	210	260
Pumping Horsepower Condenser H_2O.	20	40	40	100	80	120	60	75

C = Centrifugal. A = Absorption. Co = Continuous. Int = Intermittent.

The turbine-driven centrifugal is essentially a manual start automatic operation machine. The control system consists of a thermostat in the chilled waterline operating pneumatically the turbine governor to adjust speed of the turbine to maintain a constant leaving chilled water temperature. If the capacity continues to fall to a point where speed regulation of the turbine is beyond the scope of the governor or stability of the compressor (generally about 15 percent of full load tonnage) the speed of the turbine is maintained constant and the control automatically positions inlet guide vanes on the compressor to regulate capacity.

If it is desired to maintain the unit on the line at extremely low capacities, the standard manual hot gas bypass may be made automatic and will function below the capacity range of the turbine speed and inlet guide vane regulator. This will allow a minimum capacity for stable operation of about 5 percent of full load capacity. Generally for air-conditioning applications the speed regulation of the turbine and inlet guide vane control are sufficient.

First cost of turbine-driven centrifugals including steam condenser, cooling tower and condenser water pumps (installed) are generally lower than for absorption units.

Presented here in Table 1 is an analysis of operating data for 200, 430, 600 and 700 ton turbine-driven centrifugals as compared to the data for equivalent absorption machines. The data are taken from actual quotations and which the respec-

tive manufacturers were willing to guarantee. All comparison made on basis of 125 psig steam pressure, 53-45 F chilled water and 85 F cooling tower water.

Cost comparisons can be made utilizing local steam costs, electricity and water rates. Average figures which may be used for estimating are $0.80 per 1000 lb steam, $0.015 per kwhr and $0.60 per CF 1000 cu ft water. While these specific cases show the savings which may be attained, each individual job should be analyzed on its own merits as many variables such as steam and electric rates, water rates for specific turbines, steam pressures, and chilled water temperature desired, are involved and will be reflected in operating costs. An important analysis is part load performance and this should be made on the basis of an entire season's operation. A little ingenuity on the part of the designer will result in improvement of the heat balance by use of turbine-driven auxiliaries as well as turbine-driven chilled and condenser water pumps.

In conclusion, based upon the foregoing information it is recommended that the consulting engineer and owners investigate all of the factors which influence the selection of type of refrigeration system for their air-conditioning needs. A little extra time taken at the initial design stage may pay for itself many times over in the important consideration of operating cost which is a large part of the owning cost and in fact may at times justify a larger initial investment.

DISCUSSION

R. E. ROUSHEY, Kingston, Pa.: Can the author furnish information on the first cost per ton of a centrifugal unit including a condenser compared with an absorption unit of 200-ton capacity? The question concerns a turbine exhausting to an absorption machine.

B. H. LEE, New York, N. Y.: Can figures be given as to the relative effective operating life of the two systems?

J. J. BURKE, Philadelphia, Pa.: As a comment on the life of the centrifugal, my company has a centrifugal machine that was installed in 1927, and it is still operating as efficiently as the day it was installed and as far as can be seen there is no limit to its life. The condenser tubes and cooling tubes occasionally corrode but that is common to any form of tube.

J. C. LEE, Des Plaines, Ill.: Are the values of steam consumption for the absorption system based on a single effect or a double effect? The coefficient of performance for the double effect should be greater than for the single effect.

Does the author take into account the fact that absorption systems can be direct-fired, whereas, a steam turbine machine cannot be unless it is a gas turbine? Although not practiced, there is no reason why absorption machines cannot be built direct-fired.

W. L. BRYAN, Cleveland, Ohio: It should be pointed out that as far as corrosion is concerned in centrifugal machines it can be just as bad as in absorption machines because R11 decomposes to hydrochloric acid. If there is corrosion, the corrosion will take place at a faster rate in the absorption machine that it will in the halogenated hydrocarbon system.

AUTHOR'S CLOSURE: No direct answer to Mr. Roushey's question is possible because, first of all, my company does not produce absorption machines, and in the second place, the combined figures of the 2 are not readily available. However, in the range of 200 tons the turbine-driven system by itself, including its steam condenser, will run about $130 to $135 per ton installed. That is for the cycle only, and the absorption machine will run maybe 1 or 2 percent higher than that in the 200-hp range. Again, unit cost depends on the size turbine. In small horsepowers the same frame may be used on a

turbine for anywhere from 100 to 350 hp, but as far as the first cost comparison between absorption and centrifugal machines is concerned, the way to measure that would be to compare the absorption machine with the steam condenser since it is performing the same function, then credit for the tonnage picked up.

The best way of discussing the point concerning the relative effective life of the two systems might be to draw on experience. Some absorption machines that were introduced in 1945 and 1946 were of the open type. That is, the entire chilled water system was under vacuum. Some of those systems are still in operation but they have been extremely troublesome. Later on the closed cycle was developed and some of those have had as much as 10 years operation.

With regard to turbine-driven centrifugals some of the machines that were installed in 1936 are still operating, and with good maintenance there is every reason to believe that 25 years or even longer would be a reasonable life expectancy.

On the absorption machine the longest experience is 10 years, but probably one could expect reasonably close to the same life expectancy from these machines based upon equal maintenance.

One very important factor is that if there are too many non-condensables in the absorption system they will have a far greater effect on the life of the machine than they would on machines using halogenated refrigerants, for example, since the lithium-bromide and water and air is an extremely corrosive solution.

A question was asked about the values of steam consumption for the absorption system as affected by single or double effect. In certain cases the coefficient of performance for the double effect might be greater than for the single effect. It should be greater for the double effect than for the single effect. Performance is based on a single effect as the single effect machine is the one that is most readily available commercially at the present time.

As mentioned by Mr. Bryan, corrosion is possible in a centrifugal machine. But the corrosion effect of the lithium-bromide air and water solution is greater—a faster corrosion rate than would be experienced in a halogenated refrigerant system—unless there is a comparable amount in either system. The comment covered the normal amount of leakage in the two systems during their operating life. If the leakage exceeds the capacity of the purge unit on a centrifugal machine, the result can be a build-up in the corrosion rate, that will attack the tube. By way of an example let us say it would take 3 weeks for the corrosion to eat through the tube in the centrifugal; that time would be somewhere in the neighborhood of half if the lithium-bromide air and water solution were operating on the absorption machine tubes. The machine life in either case would be the same based upon equally good maintenance. On the other hand if there is corrosion, the corrosion will take place at a faster rate in the absorption machine than it will in the halogenated refrigerant system.

One discusser asked about direct-firing of absorption systems. That fact has not been taken into effect except that in the smaller tonnages one company is building a direct-fired absorption machine. No companies at present are offering a large direct-fired machine. In some recent advertisements of gas-fired absorption machines, down at the bottom with an asterisk, it is stated *gas-fired boiler gives you the steam for use in the absorption machine* but to the author's knowledge there are today no large tonnage direct-fired absorption machines. There may be some research or development going on right at the moment, and there is no reason why it cannot be. There may be some gain in efficiency by eliminating generation and condensing of steam. The gain might be offset somewhat by the inefficiency of the heat exchanger based upon direct-firing. Boilers are pretty efficient and it is not certain that burners and heat exchangers are any more efficient than boilers, but there might be a net gain by direct-firing.

MR. ROUSHEY: Perhaps the codes might restrict much of that direct application for steam firing.

MR. DORSEY: That may be true and probably in many cases it may be the situation.

1671

No. 1671

EFFECTS OF REFRIGERANT PROPERTIES ON CENTRIFUGAL COMPRESSOR IMPELLER DIMENSIONS AND STAGE PERFORMANCE

By F. J. WIESNER, JR.* and H. E. CASWELL**, SYRACUSE, N.Y.

CENTRIFUGAL compressors for refrigeration, especially the chilling of large volumes of water for use in relatively large air-conditioning systems of the central station type, have gained broad acceptance. The original concept (that of Dr. Willis H. Carrier, 1921) has been developed and expanded, with a multitude of improvements, to the point where refrigeration machines using centrifugal compressors are big business. The growth rate is substantial, in spite of the more recent development of such other means for chilling large quantities of water as the lithium bromide absorption refrigeration machine.

Use of centrifugal compressors in refrigeration machines has paralleled and hinged upon the simultaneous development of other machine components, and particularly has been influenced by the availability of suitable refrigerants. The heat exchangers used in association with a compressor to create a complete refrigeration machine (evaporator or cooler and condenser) are the major components so far as bulk and weight are concerned, and show high in the overall machine cost. Fig. 1, covering one line of hermetic centrifugal water chilling machines, indicates that the factory fabricated cost of heat exchangers typically varies from 30 percent to as much as 60 percent or more of the total fabricated cost (excluding the motor parts), depending upon the size of the machine and the design repercussions of the refrigerant employed. Thus, the heat exchangers deserve as much attention from the designer as does the compressor, and the design of all components is obviously inter-related. The objective must be to unify an optimal combination of components for low *first* cost, or for minimum customer *owning and operating* cost, depending upon the type of market. In general, low first cost dominates for smaller machines; owning and operating cost for larger units.

For centrifugal compressors, refrigerants should have acceptable cycle efficiencies, fairly large specific volumes, and reasonable *head* or *lift* characteristics. If they are in line as to cost, are non-toxic, will not ignite and are chemically stable, so much

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the better. In fact, the latter characteristics are virtually mandatory for acceptability in comfort air conditioning, today. A family of fluorinated hydrocarbon compounds, developed since 1921, meets the basic requirements well, yet provides considerable flexibility as to refrigeration capacity. Pertinent data for the more commonly used of these compounds are summarized in Table 1. Many good refrigerants, commonly used in the past, fail to match the inherent performance

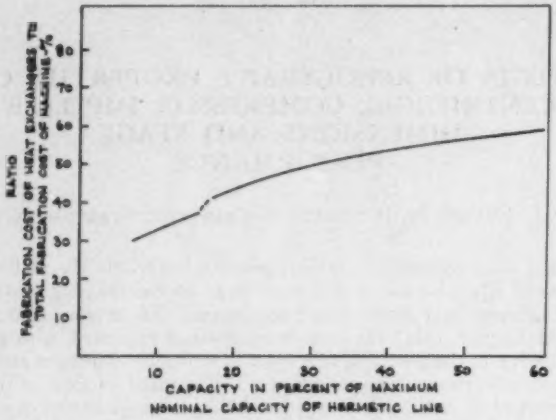

FIG. 1....RELATIVE COST OF HEAT EXCHANGERS FOR HERMETIC CENTRIFUGAL WATER CHILLING MACHINES (BASED ON FACTORY FABRICATION COSTS)

TABLE 1....COMMON FLUORINATED HYDROCARBON REFRIGERANTS USED IN CENTRIFUGAL REFRIGERATION MACHINES

FORMULA	CHEMICAL NAME	ASHRAE REFRIGERANT NO.	SATURATION PRESSURES, PSIA		SATURATED VAPOR AT 34 F	
			AT 34 F	AT 104 F	SPEC. VOLUME, CU FT PER LB	SONIC VELOCITY, fps
CCl_3F	Trichloromonofluoromethane	11	6.112	25.32	6.205	441.5
CCl_2F_2	Dichlorodifluoromethane	12	46.47	139.59	0.855	453.0
CCl_2F-$CClF_2$	Trichlorotrifluoroethane	113	2.264	11.35	12.39	372.0
$CClF_2$-$CClF_2$	Dichlorotetrafluoroethane	114	13.38	49.48	2.236	382.0
$CHClF_2$	Monochlorodifluoromethane	22	75.21	224.5	0.729	538.0
	Azeotropic Mixture of CCl_2F_2 and CH_3CHF_2	500	54.71	165.5	0.889	480.0

characteristics of centrifugal compressors. Ammonia, for example, for a water chilling temperature *lift*, requires a pumping head close to 60,000 ft. This means expensive multiple centrifugal staging, which can be avoided by using the new compounds just described.

This discussion concentrates on the effect of refrigerant choice upon centrifugal stage design, with a discussion of the various factors, including refrigerants, which influence and limit the optimum design. It will be limited to practical designs in the air-conditioning range, and will consider only those fluorinated hydrocarbon refrigerants which are most commonly used in centrifugal machines. To maintain an acceptable overall machine performance, compressor performance deficiencies must be compensated by adding heat exchanger surface—which costs money.

TABLE 2. . . .COMPRESSOR REQUIREMENTS WITH FOUR COMMONLY USED REFRIGERANTS

	SINGLE-STAGE COMPRESSOR AT CONDITIONS GIVEN IN FIG. 5				FIRST STAGE OF 2-STAGE COMPRESSOR (WITH ECONOMIZER) AT CONDITIONS GIVEN IN FIG. 6			
REFRIGERANT	11	12	113	114	11	12	113	114
Polytropic Head, ft[a]	8770	6860	7190	6030	4530	3600	3760	3200
Lb/min/ton	2.970	4.095	3.715	4.660	2.825[b]	3.785[b]	3.470[b]	4.255[b]
Cfm/ton	19.40	3.65	48.50	10.90	17.56	3.14	42.60	9.15
Cfm at 300 tons	5820	1096	14560	3270	5260	942	12800	2745

[a] At assumed 75% polytropic efficiency, but variations in this value with final efficiencies implied in Tables 3, 4, 5 and 6 are quite small.
[b] Average, including economizer effect.

Putting it another way, a substantial improvement in the compressor (usually at little or no cost) may make it possible to reduce appreciably the associated heat exchanger surface needed. Unfortunately, compensation by increasing heat exchanger surface follows the law of diminishing returns, and can quite quickly expand overall costs beyond all reason.

FACTORS AFFECTING THE PERFORMANCE

It can be demonstrated that useful output (head or pressure rise) and required power input (gas hp) for any centrifugal compressor stage, operating with any vapor, are primarily dependent upon the following variables: (a) Quantity of Flow, (b) Mach Number, (c) Reynolds Number, (d) Specific Heat Ratio, k ($= C_p/C_v$).

Useful head is expressed usually as foot pounds per pound of fluid flowing through the stage, and for the purposes of correlating data in a non-dimensional form, this quantity is commonly given as the Head or Pressure Coefficient:

$$\mu_p = \frac{gH_p}{u_2^2} \quad \ldots \ldots \ldots \ldots \quad (1)$$

The input to the gas may be expressed in many ways, but for the purposes of the development to follow, will be defined as the useful polytropic output divided by the polytropic efficiency, or non-dimensionally:

$$\mu_o = \frac{\mu_p}{\eta_p} = \frac{gH_p}{u_2^2 \eta_p} \text{ or } = \frac{gH_o}{u_2^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

As far as the quantity of flow is concerned, this is most commonly expressed in cfm at the inlet to the stage. There are also numerous non-dimensional forms for the flow quantity of centrifugal compressors. However, most investigators agree that a flow coefficient based upon the conditions of flow at the impeller tip will result in the most satisfactory correlation of data from a wide variety of stage designs and operating vapors; see Meldahl[1, 2], Sheets[3], and Stepanoff[4], among others. For the purposes of this presentation, the following dimensionless impeller tip flow coefficient has been selected:

$$\Phi_2 = \frac{c_{m2}}{u_2} \text{ (Derivation in Appendix A)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The Mach Number most commonly employed in describing the performance of centrifugal refrigeration compressors, turbochargers, and the like, is known as the Rotational Mach Number and is given as:

$$M_o = \frac{u_2}{a_o} \text{ (Dimensionless)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The Reynolds Number also most commonly used in analyses of this type is sometimes referred to as the *machine* Reynolds Number and is given as follows:

$$N_{Re} = \frac{d_2 u_2 \rho_2}{12\mu_2} \text{ (Dimensionless)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Examination of this group of equations reveals that the following vapor properties all exert some influence on the stage performance:

P-V-T characteristics of the vapor, which determine:

(a) head or pressure rise required for given temperature *lift* requirements, and

(b) density of the vapor at impeller discharge static pressure and temperature conditions.

Velocity of sound in the vapor at the stage inlet total pressure and temperature conditions.

Viscosity of the vapor at impeller tip static pressure and temperature conditions.

Specific Heat Ratio of the vapor.

The equations also indicate that the following physical dimensions (geometrical requirements) are of primary importance in connection with the stage performance:

(a) Impeller Tip Diameter, d_2, which at a given speed of rotation, N (rpm), also determines the peripheral speed of the impeller, u_2, according to:

$$u_2 = \frac{\pi d_2 N}{12 \times 60} \left(\text{Or Approx. } \frac{d_2 N}{229} \right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

(b) Impeller Tip Width, b_2, which enters into the actual determination of c_{m2}, the average radial or meridional velocity of the vapor at the impeller tip.

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[1] Exponent numerals refer to References.

NOMENCLATURE

A = flow area, square feet.
a = sonic velocity, feet per second.
b = channel width, inches.
c = absolute velocity (or components, when used with subscripts), feet per second.
C_p = specific heat of vapor at constant pressure, Btu per (pound) (Fahrenheit degree).
C_v = specific heat of vapor at constant volume, Btu per (pound) (Fahrenheit degree).
d = diameter, inches.
e = blade thickness, inches.
Ghp = gas or hydraulic horsepower, horsepower.
g = gravitational constant, 32.17 feet per (second) (second).
h = enthalpy, Btu per pound.
H = head, feet.
K = impeller meridional deceleration rate, dimensionless.
k = specific heat ratio, C_p/C_v, dimensionless.
M = Mach Number, dimensionless.
N = speed, revolutions per minute.
Q = volume flow, cubic feet per minute.
N_{Re} = Reynolds Number, dimensionless.
u = peripheral speed, feet per second.
v = specific volume of vapor, cubic feet per pound.
W = relative velocity, feet per second.
w = weight flow, pounds per minute.
Z = number of blades
β = relative flow angle (or blade angle, when there are no inlet or discharge flow deviations), with the tangential direction, degrees.
η = efficiency, expressed in %/100, dimensionless.
μ = head or pressure coefficient, dimensionless.
μ = absolute viscosity of vapor (= 0.000672 × centipoises), (feet-second) per pound.
ρ = density, pounds per cubic foot.
Φ = flow coefficient, dimensionless.

Subscripts:

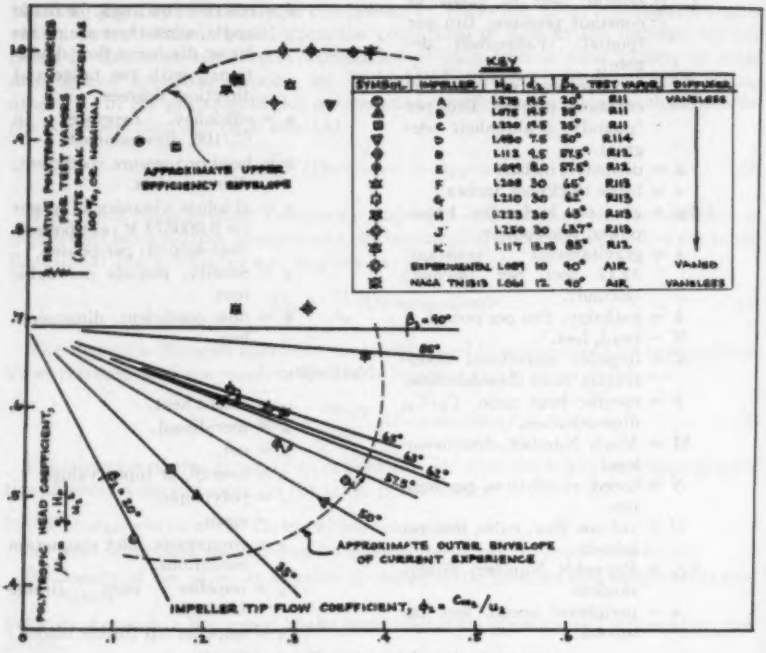
$_{lg}$ = latent heat.
$_m$ = meridional.
$_n$ = net.
$_o$ = overall, or input values.
$_p$ = polytropic.
$_s$ = static.
$_0$ = compressor inlet stagnation conditions.
$_1$ = impeller inlet (inside blades).
$_2$ = impeller tip (inside blades.)

There are a few other secondary geometrical requirements or limitations for the stage, among which are the impeller tip blade angle and the maximum inlet relative Mach Number, which enter into the ability of the design to produce *peak* performance. These considerations will be discussed later, in connection with some specific examples.

TYPICAL PEAK STAGE PERFORMANCE CHARTS

As a result of the correlation of a large amount of test data, similar to that shown in Fig. 2, obtained from centrifugal stages of many different designs (operating in a wide variety of vapors), taken in conjunction with other data which are available readily [principally Meldahl[2], Sheets[3], Balje[5], Wosika[6] and Stepanoff[4]], the diagrams

shown in Figs. 3 and 4 were constructed. These charts are felt to describe adequately the practical obtainable peak performance of any carefully designed centrifugal compressor stage, operating with any vapor. Thus, any point on Fig. 3 may be considered to be a *design point* (for no inlet pre-rotation), and it should be noted that the predominant effects of the Flow Coefficient, Impeller Tip Blade Angle, and Rotational Mach Number on both the Polytropic Head Coefficient and



(Impellers A to K are those of the authors' company.)

FIG. 2. . . .TEST PERFORMANCE CORRELATIONS (AT PEAK EFFICIENCIES) IN DIMENSIONLESS COORDINATES

the Polytropic Efficiency may all be represented on a single chart. Here, it should be pointed out that the test polytropic efficiencies shown in Fig. 2 have been *normalized* and subsequently termed Relative Polytropic Efficiencies. This was accomplished by taking the absolute value of the peak polytropic efficiencies for the 4 best efficiency test points, (Impellers E and K, and the Experimental Impeller, all at $M_o \cong 1.10$), as a nominal (or 100 percent) value, and expressing all other test efficiencies in terms of this nominal.

The Relative Polytropic Efficiencies shown in Fig. 3 are also expressed in the same terms, and it can be seen that the nominal values of Figs. 2 and 3 are in agree-

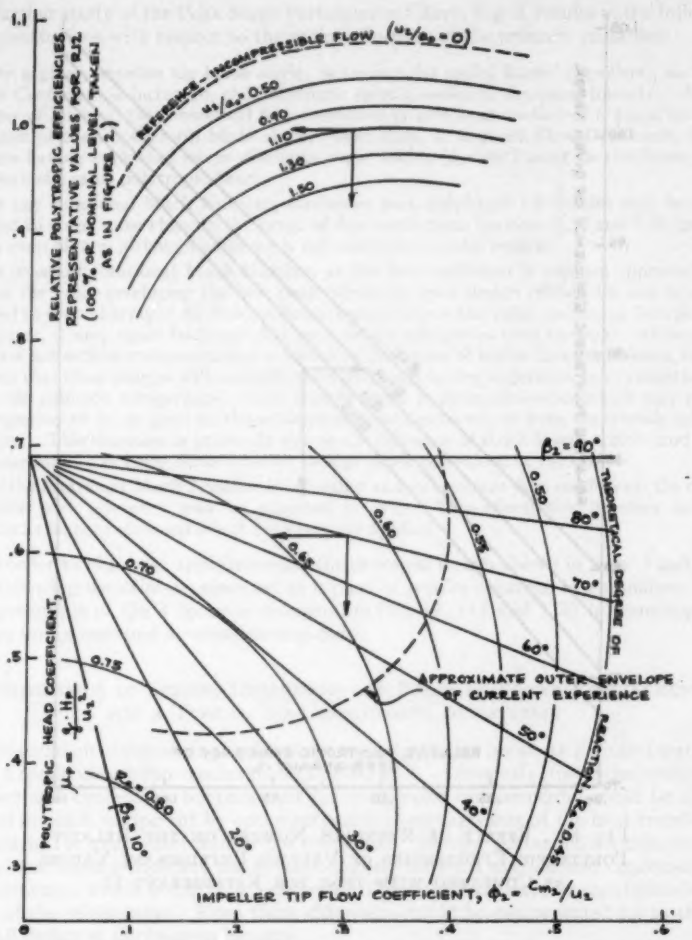


FIG. 3. . . .TYPICAL PEAK STAGE PERFORMANCE CHART (DIMENSIONLESS COORDINATES)

ment at a Rotational Mach Number of about 1.10 in the region of high flow coefficients. Fig. 4 indicates the slight (but not completely negligible) influence of Reynolds Number on the Polytropic Efficiency for several of the more common refrigerants. Note that the relative efficiencies shown in Fig. 3 are specifically labeled as representative of those attainable with Refrigerant 12, since this vapor

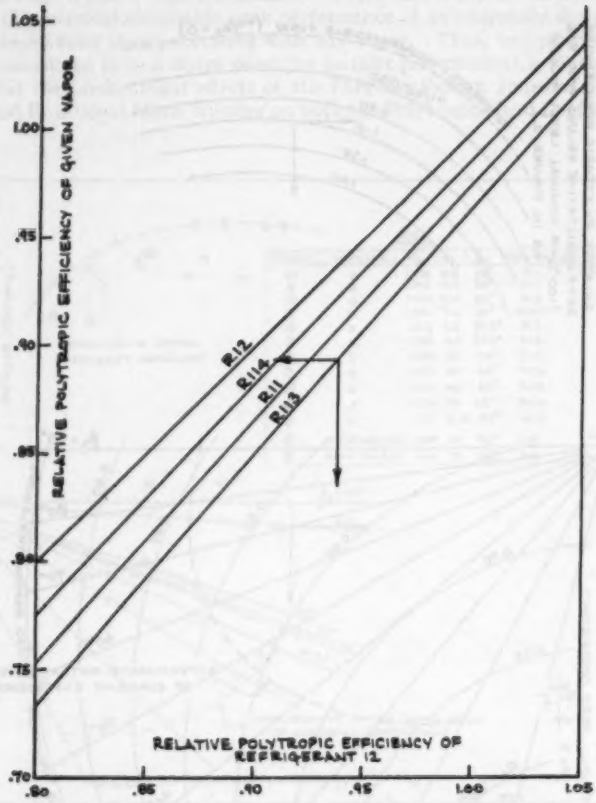


FIG 4. . . .EFFECT OF REYNOLDS NUMBER ON THE RELATIVE POLYTROPIC EFFICIENCIES OF VARIOUS REFRIGERANT VAPORS AS COMPARED WITH THAT FOR REFRIGERANT 12

has the highest Reynolds Number among the 4 most common refrigerants (11, 12, 113 and 114), used in centrifugal air-conditioning machines. With reference to Fig. 4, the rest of these refrigerants would be expected to rank in descending order as follows: Refrigerants 114, 11 and 113. Of the higher tonnage refrigerants, Refrigerant 500 is expected to be equal to Refrigerant 12, while Refrigerant 22 may be considered slightly better as a result of even higher Reynolds Numbers in the air-conditioning range of applications.

Again referring to Fig. 4, Relative Polytropic Efficiencies are shown in place of the absolute values, with the reference value of the polytropic efficiency of Refrigerant 12 being chosen to coincide with that used in Figs. 2 and 3.

Further study of the Peak Stage Performance Chart, Fig. 3, results in the following conclusions with respect to the main influences of the primary variables:

For a given impeller tip blade angle, β_2 (except for radial bladed impellers), as the Flow Coefficient is increased, the Polytropic Head Coefficient decreases linearly. As a matter of interest, the theoretical flow coefficient at zero head coefficient is equal to the tangent of the impeller tip blade angle. Obviously, at a given Flow Coefficient, impellers having backward swept discharge vane angles ($\beta_2 < 90°$) must be run faster to produce the same polytropic head.

At any rotational Mach Number, maximum peak polytropic efficiencies may be expected to occur somewhere in the range of flow coefficients between 0.30 and 0.35 (perhaps even higher, although evidence is not conclusive in this regard).

At constant rotational Mach Number, as the flow coefficient is reduced appreciably below the value producing the best peak efficiency, peak design efficiencies can be expected to drop sharply. At flow coefficients much above the value producing best peak efficiency, it may again be found that peak design efficiencies tend to drop. Although there is not sufficient experimental evidence in the region of higher flow coefficients, it is known that these designs will normally produce stages having supersonic inlet conditions with the common refrigerants. Such designs result in peak efficiencies which may not be expected to be as good as those obtainable with subsonic or even trans-sonic inlet designs. This situation is primarily due to the influence of shock losses, which tend to increase rapidly in going from subsonic to supersonic conditions at the inlet.

As the rotational Mach Number is increased at any constant flow coefficient, the obtainable peak efficiency may be expected to drop. This effect, too, becomes more serious as higher rotational Mach Numbers are applied.

In order to illustrate applications of the practical trends shown in Figs. 3 and 4, the following material is presented as typical of results obtained in an analysis of the potentials of the 4 common refrigerants (11, 12, 113 and 114) in centrifugal stages for an assumed air-conditioning duty.

DETERMINATION OF SPEEDS, DIMENSIONS AND RELATIVE POWER REQUIREMENTS FOR A TYPICAL AIR-CONDITIONING APPLICATION

The usual air-conditioning design requirement calls for about 44 F chilled water with a condensing water temperature of about 85 F. Obviously, the corresponding suction and condensing temperatures (or compressor requirements) could be obtained for each refrigerant by optimum practical arrangement of the heat transfer surface in the evaporator and in the condenser. Having accomplished this, it is likely that each refrigerant would have slightly different suction and condensing temperatures, which would be a direct reflection of the heat transfer capabilities of each of the refrigerants. Even these differences could be compensated for in the overall design of the heat exchangers.

Since it is outside of the scope of this paper to consider the heat transfer aspect of the problem, one must assume some typical corresponding suction and condensing temperatures which will be held constant for all of the refrigerants to be analyzed. These temperatures are most commonly found to be about 34 F in the evaporator and about 104 F in the condenser, for the water conditions just given. The assumed suction and condensing temperatures, together with the physical properties of each refrigerant, and the assumed standard cycles shown in Figs. 5 and 6, result in the compressor requirements listed in Table 2 for Refrigerants 11, 12, 113 and 114. Although these analyses can be undertaken for any load, a typical load at 300 ton of refrigeration has been chosen. Note the appreciable variations in the head,

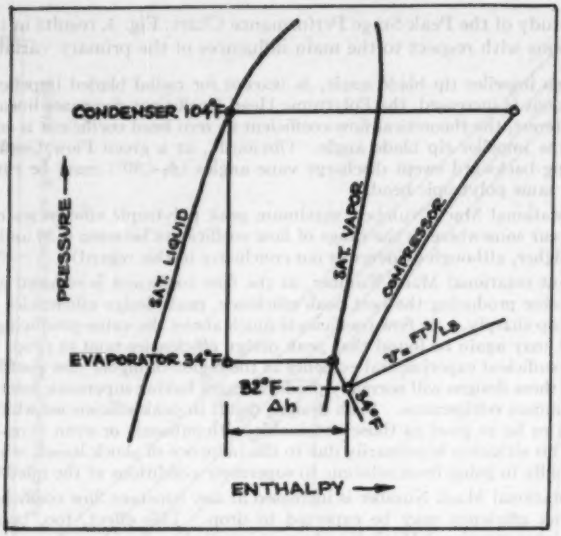


FIG. 5. . . .TYPICAL REFRIGERATION CYCLE ASSUMED FOR SINGLE STAGE UNITS

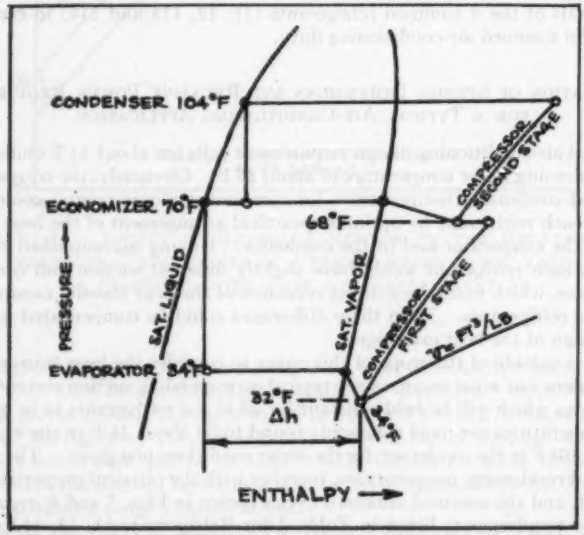


FIG. 6. . . .TYPICAL ECONOMIZING REFRIGERATION CYCLE ASSUMED FOR TWO STAGE UNITS

weight flow, and suction volume requirements for the 4 refrigerants at the constant load and temperature *lift* requirements.

Before entering into the determination of compressor dimensions, speeds and relative power requirements, some further general assumptions must be made with respect to the compressors as follows:

Maximum Permissible Inlet Relative Mach Number: (at the inducer or impeller eye tip) usually set at 0.90 for the highest flow compressor in any given *family* of machines, although a more conservative design would limit this value to 0.80 or less.

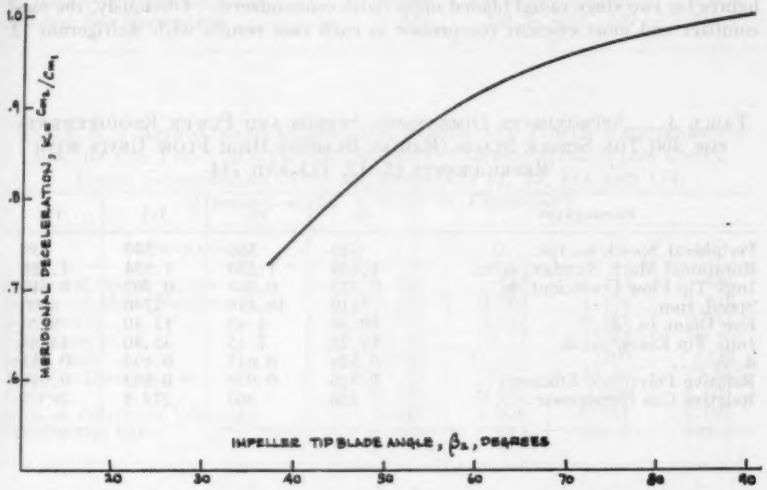


FIG. 7. . . .APPROXIMATE ALLOWABLE MERIDIONAL DECELERATION VS IMPELLER TIP BLADE ANGLE

Inlet Prerotation: The information of Fig. 3 is intended for stages having no inlet prerotation, and must be adjusted when either with rotation or against rotation velocity components are induced at the impeller inlet (usually with the aid of fixed or variable inlet guide vanes, although prerotation can be *self induced* by improper inlet designs).

Optimum Inlet Flow Triangle (at the inducer tip): This can be shown to vary principally with the assumed Inlet Relative Mach number and the specific heat ratio of the vapor. Over a wide range of these variables, however, the optimum inlet angle varies only a small amount from an average value of 32° with the tangent.

Inlet Free Area Factor: In order to size the impeller eye or inducer tip diameter, one must assume some average restrictive effect of the impeller hub and blades. Also, there is a certain amount of expansion of the vapor as it flows from the evaporator to the impeller inlet. A factor of 82.5 percent free area may be assumed as representative of a typical practical design. This factor must be carefully evaluated for each specific final stage design.

Allowable Deceleration Rate of the Radial or Meridional Velocities from Impeller Inlet to Outlet: This is commonly specified at $K = c_{m2}/c_{m1}$ = no less than 1.00 for radial bladed impellers, ($\beta_2 = 90°$). For impellers with backward swept discharge blade angles, several investigators agree that a certain amount of meridional deceleration is permissible (for the same safe margin against flow separation in the impeller). Fig. 7 includes some representative values for allowable deceleration rates for backward swept impeller discharge blade angles, as a function of the angle β_2.

Taking these considerations or limitations into account, one may calculate the necessary stage dimensions, speeds, and relative power requirements for all of the common refrigerants under the conditions given in Table 2. Table 3 shows the results for single stage radial bladed units, while Table 4 presents comparable figures for two stage radial bladed units (with economizers). Obviously, the most compact and most efficient compressor in each case results with Refrigerant 12.

TABLE 3. . . .APPROXIMATE DIMENSIONS, SPEEDS AND POWER REQUIREMENTS FOR 300 TON SINGLE STAGE (RADIAL BLADED) HIGH FLOW UNITS WITH REFRIGERANTS 11, 12, 113 AND 114

REFRIGERANT	11	12	113	114
Peripheral Speed, u_2, fps	640	566	579	530
Rotational Mach Number, u_2/a_o	1.450	1.250	1.554	1.388
Imp. Tip Flow Coefficient, Φ_2	0.323	0.369	0.302	0.340
Speed, rpm	7610	18,170	3730	8190
Eye Diam, in., d_e	10.20	4.40	17.50	8.20
Imp. Tip Diam, in., d_2	19.28	7.15	35.50	14.83
d_e/d_2	0.529	0.615	0.493	0.553
Relative Polytropic Efficiency	0.926	0.979	0.893	0.949
Relative Gas Horsepower	256	261	272.5	269.5

The largest and least efficient compressor occurs with the choice of Refrigerant 113. Refrigerants 11 and 114 are nearly the same, and intermediate in these respects. When consideration is taken of the overall cycle performance, as represented by the Relative Gas Hp criterion, it can be seen that Refrigerant 11 should be expected to produce a somewhat better machine, followed by Refrigerants 12, 114 and 113, respectively. For machines operating in economizing cycles under the conditions assumed, it can be noted that the overall relative performance with the latter three refrigerants are all nearly identical (about 239 Relative Gas Hp).

The impeller tip flow coefficients for the single stage units listed in Table 3 all fall within the envelope of current experience shown in Fig. 3, however, the corresponding flow coefficients for the two-stage units shown in Table 4 are all beyond the limits of current experience, and may be expected to present some critical design and performance problems. It should be mentioned here that the relative efficiency trends of Fig. 3 have been extrapolated in this region.

In order to bring the designs of Table 4 into the region of current experience, while maintaining radial bladed impellers, one could limit the impeller tip flow coefficient to a maximum value of $\phi_2 = 0.375$ for all refrigerants. In this case, the results shown in Table 5 would be obtained. Note that under these conditions,

TABLE 4. . . .APPROXIMATE DIMENSIONS, SPEEDS AND POWER REQUIREMENTS FOR 300 TON TWO STAGE (RADIAL BLADED) HIGH FLOW UNITS, USING ECONOMIZERS WITH REFRIGERANTS 11, 12, 113 AND 114

REFRIGERANT	11	12	113	114
Peripheral Speed, u_2, fps	460	410	419	386
Rotational Mach Number, u_2/a_o	1.040	0.905	1.124	1.010
Imp. Tip Flow Coefficient, Φ_2	0.450	0.510	0.418	0.466
Speed, rpm	8000	19,570	3960	8910
Eye Diam, in., d_e	9.70	4.09	16.50	7.53
Imp. Tip Diam, in., d_2	13.19	4.80	24.20	9.93
d_e/d_2	0.735	0.851	0.681	0.758
Relative Polytropic Efficiency	0.968	0.992	0.951	0.979
Relative Gas Horsepower	233	238.5	239	238.7

TABLE 5. . . .APPROXIMATE DIMENSIONS, SPEEDS AND POWER REQUIREMENTS FOR 300 TON TWO STAGE (RADIAL BLADED) "CONSERVATIVE" FLOW UNITS, USING ECONOMIZERS, WITH REFRIGERANTS 11, 12, 113 AND 114. (DESIGNS AT $\Phi_2 = 0.375$ = CONSTANT)

REFRIGERANT	11	12	113	114
Peripheral Speed, u_2, fps	460	410	419	386
Rotational Mach Number, u_2/a_o	1.040	0.905	1.124	1.010
M_1, relative	0.737	0.642	0.795	0.717
Eye Diam, in., d_e	10.60	4.77	17.45	8.40
Speed, rpm	5960	11,870	3300	6330
Imp. Tip Diam, in., d_2	17.72	7.92	29.05	13.96
d_e/d_2	0.598	0.602	0.600	0.601
Relative Polytropic Efficiency	0.980	1.018	0.957	0.995
Relative Gas Hp	230	233	237.5	234.5

TABLE 6. . . .APPROXIMATE DIMENSIONS, SPEEDS AND POWER REQUIREMENTS FOR 300 TON TWO STAGE HIGH FLOW UNITS (WITH BACKWARD SWEPT IMPELLER BLADING), USING ECONOMIZERS, WITH REFRIGERANTS 11, 12, 113 AND 114. (DESIGNS JUST AT LIMITS OF CURRENT EXPERIENCE)

REFRIGERANT	11	12	113	114
Peripheral Speed, u_2, fps	508	486	435	428
Rotational Mach Number, u_2/a_o	1.149	1.073	1.169	1.120
Imp. Tip Flow Coefficient, Φ_2	0.390	0.345	0.395	0.395
Imp. Tip Blade Angle, β_2	65°	50°	80°	65°
Speed, rpm	8000	17,180	3960	8910
M_1, Relative	0.90	0.823	0.90	0.90
Eye Diam, in., d_e	9.70	4.21	16.50	7.53
Imp. Tip Diam, in., d_2	14.57	6.49	25.17	11.00
d_e/d_2	0.666	0.649	0.655	0.685
Relative Polytropic Efficiency	0.966	1.002	0.950	0.979
Relative Gas HP	233.5	236	239.5	238.5

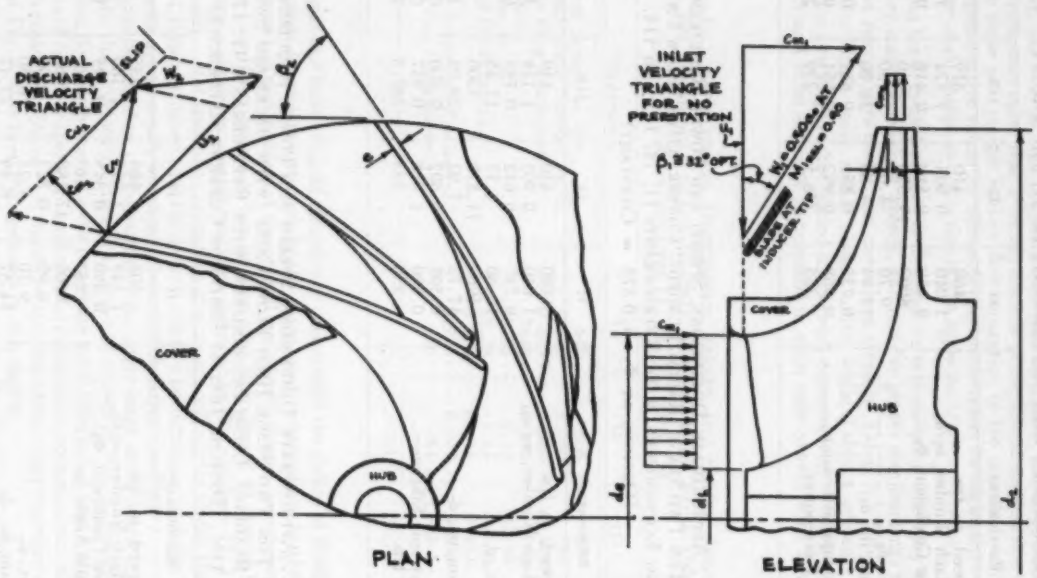


FIG. 8. . . .SIGNIFICANT IMPELLER DIMENSIONS AND INLET AND DISCHARGE VELOCITY TRIANGLES

the relative inlet Mach Numbers have been materially reduced, the rotational speeds have been decreased (with corresponding diametral increases), and the stages have moved into a region of performance where the efficiencies may be expected to improve. Again, on a relative gas hp basis, Refrigerant 11 is still best, followed by Refrigerants 12, 114 and 113, respectively.

The designs of Table 4 may be brought into the region of current experience without changing the inlet relative Mach Numbers (except for Refrigerant 12) by considering backward swept impeller blading, at the same time allowing some nominal amounts of meridional deceleration as suggested by Fig. 7. In this case, the values shown in Table 6 are typical of the result. These stages are somewhat larger than those in Table 4, but somewhat smaller than those in Table 5. The relative efficiencies and corresponding power values are, however, comparable to those in Table 4, except for Refrigerant 12, where the inlet relative Mach Number was further limited (to 0.823, only to produce a design within the current range of experience).

The figures in Tables 3 and 6 would be representative of the final overall impeller dimensions, speeds and relative power requirements for high flow single-stage and two-stage (with economizer) machines for Refrigerants 11, 12, 113, and 114, at 300 ton and 34 F evaporator and 104 F condensing temperatures. Similar analyses are entirely possible with the aid of these charts, at any other load and lift requirements for any vapor.

Professor Soo[7] has presented a similar analysis, where he assumed a pressure coefficient of 0.70, and an overall adiabatic compressor efficiency of 78 percent (the same for all refrigerants). His results were presented for two-stage units, with economizers, at 0.90 inducer tip Mach Numbers and operating between 38 and 105 F. Without so stating, his assumption of 0.70 pressure coefficient obviously limits the application of his results to radial bladed stages only. Also, as with Table 4, the impeller dimensions and speeds given in Professor Soo's Figs. 1, 2 and 3 are considerably beyond the envelope of current experience. So far, these assumptions do not violate the basic intents of his paper; *i.e.*, the presentation of a sequence of maximum design capacities for various refrigerants at various speeds and inducer tip Mach Numbers. In assuming a constant compressor efficiency for all of these various possibilities, however, Professor Soo has failed to complete the story and indicate the predominant effects of flow coefficient, Rotational Mach Number and Reynolds Number on the relative efficiency levels or relative power requirements of the various vapors. This could be done now with the aid of Fig. 3, by projecting the polytropic equivalent of the 78 percent adiabatic efficiency, assumed by Professor Soo, into the framework of this chart at his flow coefficient and Rotational Mach Number for (say) Refrigerant 12. The corresponding relative polytropic efficiency of Fig. 3 consequently takes on this assumed absolute value, and all other points on the chart are then defined in terms of this value. In like manner, absolute values can be assigned within the framework of Fig. 4. When this is accounted for, along with the other conditions assumed by Professor Soo, the comparable adiabatic efficiencies which may be expected for the 4 most common refrigerants would be similar to the following:

Refrigerant 11—76.6 percent
Refrigerant 12—78.0 percent (the assigned value)
Refrigerant 113—75.4 percent
Refrigerant 114—77.6 percent

In addition to allowing an analysis to be made for any impeller tip blade angle, the material presented in this paper will allow evaluations of designs which do not violate the current practical range of experience, and at the same time will distinguish between vapors as far as obtainable compressor efficiencies are concerned.

SUMMARY

It is felt that this paper presents that significant material necessary to specify basic impeller design dimensions and speeds of any centrifugal impeller, for use with any refrigerant vapor. At the same time, it is also possible to take advantage of those features which will govern the attainment of the best peak stage efficiencies. Obviously, as far as the compressor is concerned, the best efficiency will produce the least gas horsepower per ton of refrigerating effect. It is assumed that all major properties which tend to influence the peak efficiency of any stage, with any vapor, are adequately described, and represented in Figs. 3 and 4 of this paper. The relative polytropic efficiencies shown on these plots may be converted to representative absolute values by assuming a practical nominal value for some point within the field of Fig. 3 (as mentioned in connection with the discussion of the paper presented by Professor Soo).

With respect to the 300 ton comparisons calculated for Refrigerants 11, 12, 113, and 114, and presented in Tables 3 through 6 of the paper, one may summarize the results as follows:

The most compact and most efficient compressor results with the use of Refrigerant 12.

In consideration of the overall machine performance, Refrigerant 11, with a somewhat poorer compressor efficiency, realizes the best gas hp per ton of refrigeration.

The largest and least efficient compressor is obtained with Refrigerant 113. Also, this refrigerant is the poorest on a gas hp per ton basis.

In the economizing cycle chosen for these comparisons, Refrigerant 11 still realizes the best power per ton, with savings of the order of 9 to 10 percent over the single-stage cycle. Similar savings for the other refrigerants are found to be approximately as follows:

Refrigerant 12— 9–11 percent
Refrigerant 113—14–15 percent
Refrigerant 114—13–15 percent

With regard to speeds at this load, Refrigerant 113 is most adaptable for direct motor drive, while the other refrigerants would require either speed increasing elements or high speed turbines, and possibly even both of these media in the case of Refrigerant 12. Obviously, it is this advantage of Refrigerant 113 which results in its use for this load range in direct drive hermetic compressors, in spite of the fact that its overall performance is not as good as that of the other refrigerants.

The charts presented in this paper should be most useful in correlating or evaluating the design considerations and the relative performance abilities of newer refrigerants as they come into existence for practical application in centrifugal refrigeration compressors.

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APPENDIX A

RELATIONSHIP BETWEEN INLET VOLUME FLOW AND IMPELLER TIP FLOW COEFFICIENT

The inlet volume flow, Q_o, usually expressed in cfm is the normally measured flow quantity of centrifugal compressors (or else, the weight flow and specific volume are obtained from conditions in the evaporator, from which $Q_o = w\, v_o$). Since the weight flow remains a constant through the stage, one may write the impeller tip volume as follows:

$$Q_2 = wv_2$$

hence the tip volume in terms of the inlet volume may be written

$$Q_2 = Q_o \frac{v_2}{v_o}$$

Now, it can be shown that the volume ratio, v_o/v_2 or its inverse, the density ratio, ρ_2/ρ_o, relating inlet conditions to the static conditions at the tip of the impeller, is dependent upon the specific heat ratio of the vapor (k), the impeller tip static head coefficient, and the Rotational Mach Number, in an adiabatic process, as follows:

$$\rho_2/\rho_o = [1 + (k-1)\, \mu_o\, M_o^2]^{1/(k-1)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-1)}$$

Since most well designed impellers operate at notably high efficiency (sometimes reaching well above 90 percent), the process through this portion of the stage is nearly adiabatic. Also, a close approach to the impeller tip static head coefficient may be obtained by multiplying the useful overall stage head coefficient, μ_p, by a factor known as the *theoretical degree of reaction*, R_a, which is an estimate of the amount or percent of the stage head realized as static rise in the impeller. As shown in Fig. 3, this quantity is generally high for backward swept impeller blading, and is relatively low for radial bladed impellers. Including these quantities in the equation for density ratio, the following is obtained.

$$\rho_2/\rho_o = [1 + (k-1)\, \mu_p R_a M_o^2]^{1/(k-1)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-2)}$$

Now express the impeller tip volume as follows:

$$Q_2 = Q_o/[1+(k\text{-}1)\mu_p R_a M_o^2]^{1/(k-1)}$$

Also, in terms of the dimensions of the impeller and its outlet radial velocity, the tip volume is also equal to:

$$Q_2 = 60\, c_{m2} A_2 \quad \text{(A-3)}$$

where A_2 is taken as the net tip area, or

$$A_2 = \frac{\pi d_2 b_2}{144} - \frac{Z b_2 e}{144 \sin \beta_2} \quad \text{(A-4)}$$

The second of these terms is the amount of the total area taken up by the blades, and for a great majority of impellers, this amounts to only 5 to 8 percent of the total projected area. On the average, one could write:

$$A_2 \cong \frac{0.93\,\pi d_2 b_2}{144}, \text{ or simply } \frac{d_2 b_2}{49.3} \quad \text{(A-5)}$$

In any event, one may write the following:

$$Q_2 = 60\, c_{m2} A_2 = Q_o/[1+(k\text{-}1)\mu_p R_a M_o^2]^{1/(k-1)}$$

or

$$c_{m2} = Q_o/60 A_2[1+(k\text{-}1)\mu_p R_a M_o^2]^{1/(k-1)}$$

and

$$\Phi_2 = \frac{c_{m2}}{u_2} = Q_o/60 u_2 A_2[1+(k\text{-}1)\mu_p R_a M_o^2]^{1/(k-1)} \quad \text{(A-6)}$$

which is the quantity and form in which this study is interested.

APPENDIX B

SAMPLE CALCULATIONS

Example: Using the data given in Figs. 3, 4 and 7 and Table 2 of this paper, determine the dimensions, speeds and relative power requirements for 300 ton Refrigerant 113 centrifugal refrigeration compressors as follows:
(a) Single stage with radial blades, at a maximum inlet relative Mach Number of 0.90.
(b) Two stage (with economizer), with backward swept blading (just sufficient to place the design on the "envelope of current experience"), at a maximum inlet relative Mach Number of 0.90.

Solution for (a) single stage with radial blades: For the cycle shown in Fig. 5 of the paper, a Refrigerant 113 compressor will be required to handle 14,560 cfm and produce a polytropic head of about 7190 ft. (Table 2). At an optimum inducer tip flow angle of about 32°, and an inlet relative Mach Number of 0.90, the impeller blade relative velocity = 0.90 × 372 (sound velocity) = 334 ft/sec, the inlet meridional velocity, $c_{m1} \cong 334 \times \sin 32° = 175$ ft/sec and the inducer tip peripheral speed = 334 × cos 32° ≅ 285 ft/sec, for no inlet prerotation. The net inlet area required for this design is

$$A_1 = \frac{14{,}560}{60 \times 175} = 1.387 \text{ sq ft } (c_{m1} \text{ assumed uniform})$$

Allowing a free area factor of 0.825, accounting for the impeller hub and blade blockage effects, and for some expansion of the vapor from the evaporator to the impeller inlet, the required total impeller eye area becomes:

$$A_e = \frac{1.387 \times 144}{0.825} = 242 \text{ sq in.}$$

which results in an impeller eye diam of about 17.50 in. This dimension, taken with the optimum peripheral speed required at this diameter, $u_1 = 285$ ft/sec, establishes the rotational speed in rpm:

$$N = \frac{285 \times 12 \times 60}{\pi \times 17.5} = 3730 \text{ rpm}$$

Referring now to Fig. 3, the polytropic head coefficient is a constant and equal to 0.69 for radial bladed impellers, $\beta_2 = 90°$ regardless of the flow coefficient. One may therefore determine the peripheral speed of the impeller from Equation 1 of the paper as follows:

$$0.69 = \frac{32.17 \times 7190}{u_2^2}, \quad \text{or} \quad u_2 = 579 \text{ ft/sec}$$

which establishes the Rotational Mach Number $M_o = u_2/a_o = 579/372 = 1.554$, and the impeller tip diameter:

$$d_2 = \frac{579 \times 12 \times 60}{\pi \times 3730} \cong 35.50 \text{ in.}$$

The Impeller Tip Flow Coefficient $\Phi_2 = c_{m2}/u_2 = c_{m1}/u_2$ for this radial bladed design since one should prescribe $K = c_{m2}/c_{m1} =$ no less than 1.00 according to Fig. 7. Thus, the impeller tip flow coefficient becomes

$$\Phi_2 = \frac{175}{579} \cong 0.303,$$

which is certainly within the range of experience, according to Fig. 3. At this flow coefficient, and a Rotational Mach Number of 1.554, Fig. 3 indicates a relative polytropic efficiency of about 0.938 for Refrigerant 12. A Reynolds Number correction for Refrigerant 113 may be obtained from Fig. 4 by entering the chart on the bottom at a value of 0.839, then proceeding upward to the Refrigerant 113 line, and finally reading the corresponding relative polytropic efficiency of 0.893, at the left hand side of this figure. A relative gas horsepower may now be calculated as follows:

$$\text{Ghp} = \frac{\text{lb/min/ton} \times \text{ton} \times \text{head}}{33000 \times \text{rel. polytropic efficiency}} = \frac{3.715 \times 300 \times 7190}{33000 \times 0.893} = 272.5 \text{ hp}$$

Solution for (b) Two stage (economizing) with backward swept blades: This solution proceeds in a manner similar to the previous solution, except here a point is found on the *envelope of experience* in Fig. 3, which can simultaneously satisfy both of the following conditions:

An inlet relative Mach Number of 0.90, with an optimum flow angle of about 32°, and
The maximum allowable deceleration rates given in Fig. 7.

This may be done quickly by trial and error, with the result that an impeller tip blade angle of 80° is found to be the maximum permissible. The (Fig. 3) impeller tip flow coefficient is 0.395 and the polytropic head coefficient is 0.64. This time, the required impeller tip speed is:

$$u_2 = \sqrt{\frac{32.17 \times 3760}{0.64}} = 435 \text{ ft/sec,}$$

and the Rotational Mach Number is, $M_o = 435/372 = 1.169$.
With $c_{m2}/u_2 = 0.395$, $c_{m2} \cong 172$ ft/sec. Now Fig. 7 at an impeller tip blade angle of 80° indicates an allowable value of $c_{m2}/c_{m1} \cong 0.983$. Therefore, $c_{m1} = 175$ ft/sec (the

same as for the single stage inlet—thus producing the same inlet velocity triangle, and inlet relative Mach Number as before). This time the net inlet area becomes:

$$A_1 = \frac{12800}{60 \times 175} = 1.220 \text{ sq ft}$$

Again, with the same free area factor, the eye area becomes 213 sq in. and the eye diam is about 16.50 in.

The rotational speed in this case becomes:

$$N = \frac{285 \times 12 \times 60}{\pi \times 16.5} = 3960 \text{ rpm},$$

and one may also calculate the impeller tip diameter:

$$d_2 = \frac{435 \times 12 \times 60}{\pi \times 3960} = 25.20 \text{ in.}$$

Referring again to Fig. 3 at $\Phi_2 = 0.395$ and $u_2/a_o = 1.169$, the relative polytropic efficiency, if Refrigerant 12 were the vapor, would be approximately 0.987. With the correction for Reynolds Number applied in this case, one may expect the relative polytropic efficiency of Refrigerant 113 to be about 0.950.

The corresponding relative gas horsepower for this stage (alone) can be calculated as follows:

$$\text{Ghp} = \frac{w \times H_p}{33000 \times \eta_p} = \frac{3.26 \times 300 \times 3760}{33000 \times 0.950} = 116.3 \text{ hp}$$

however, this cannot be compared directly with the overall hp of the single stage unit of *Solution* (*a*).

In order to arrive at more comparable figures (without going through an analysis of the second stage for this machine), the following assumptions have been made:

The mean weight flow for the two stages is equal to the first stage weight flow plus one half of the economizer weight flow.

The overall two-stage polytropic head is the same as the head required for the single stage machine.

The efficiency of the entire two stage machine is the same as the efficiency of the first stage alone.

It can be demonstrated that these assumptions are nearly correct. In any event, the relative gas horsepower of the two stage unit, with economizer, in accordance with these assumptions, becomes:

$$\text{Ghp} = \frac{3.26 \times 1.063 \times 300 \times 7190}{33000 \times 0.950} = 239.5 \text{ hp}$$

The results of *Solution* (*a*) are given in Table 3 of the paper, while the results of *Solution* (*b*) are given in Table 6. The results for all of the other refrigerants and conditions shown in Tables 3 to 6 were calculated in the same manner.

DISCUSSION

W. G. DORSEY, Kansas City, Mo.: Did the authors find any variances among refrigerants under partial load performance conditions?

AUTHORS' CLOSURE (Mr. Wiesner): The study included only the investigation of peak performance conditions, hence the analysis of the common refrigerants presented here is specifically applicable to these conditions. Mr. Dorsey has indicated another area which should also be investigated, but we have not done so, and therefore, have no data to present. It is this author's feeling, however, that virtually the same variances would be found between refrigerants under partial load performance conditions that were found under peak performance conditions, and the behavior would be generally the same regardless of whether the partial load conditions were met by means of a change in speed or with a change in guide vane position.

J. C. LEE, Des Plaines, Ill.: Is it correct that the head is expressed in terms of feet of inlet condition?

AUTHORS' CLOSURE (Mr. Wiesner): The head used in this presentation is the useful overall polytropic head for the complete stage, determined from the product of the log of the overall stage pressure ratio and the logarithmic mean pv across the stage, or

$$H_p = \ln \frac{p_2}{p_1} \left[\frac{p_2 v_2 - p_1 v_1}{\ln \frac{p_2 v_2}{p_1 v_1}} \right]$$

Thus, both the inlet and discharge conditions of the vapor enter into the head calculation.

J. C. LEE, Des Plaines, Ill.: Is that equivalent to the enthalpy change across the impeller?

AUTHORS' CLOSURE (Mr. Wiesner): No, the head equivalent of the enthalpy rise across the impeller which the authors term the input head (see Equation 2 of the paper), is equal to the overall polytropic head divided by the overall polytropic efficiency. In the authors' test work they measured the pressure rise and the temperature rise across the stage, and these data together with the p-v-t properties of the vapor are sufficient to calculate both heads. The overall stage efficiency is then determined from the relation

$$\eta_p = \frac{H_p}{H_e}$$

There appears to be a little difficulty in understanding the relationship between the useful overall stage performance, which the authors have presented in generalized form, and the performance of the impeller by itself. As far as the overall performance is concerned, this is certainly dependent upon all of the elements which make up a complete stage. These elements normally include the inlet pipe, inlet guide vanes, impeller, vaneless diffuser, and return channel or volute. In order to achieve peak performance, each of these elements must be designed to match properly at given flow conditions. Among these elements, however, the impeller is of primary importance because it is the only component which transmits the shaft power to the vapor. All of the other elements are really nothing more than power loss devices which are necessary to guide the flow and/or recover as much of the kinetic energy (supplied by the impeller) as possible. At peak performance for well matched designs, the stationary elements apparently always behave in such a way that the impeller configuration becomes a good criterion of the practical overall stage performance. This does not mean that designing the stage in different ways around the same impeller will always yield the same overall stage performance, but there is apparently a considerable amount of leeway in the design of most of the stationary elements which will yield almost the same overall performance with a given impeller. It is this tendency which allows us to get satisfactory practical overall performance correlations, simply related to the basic impeller dimensions. Those special designs which tend to produce exceptional overall performance are usually found

to be impractical on the basis of economics (for the classes of machinery under discussion here), because they normally can be expected to add to the costs of production.

S. P. SOLING, York, Pa.: In Figs. 5 and 6 there is shown a *net latent heat* extending from a point in the wet region, that is between the saturated liquid and saturated vapor lines, to a point in the superheat region. Since latent heat is usually taken as the change in enthalpy between the saturated liquid and the saturated vapor lines, we believe that the use of a *net latent heat* can be confusing. This comment is offered in the hope that in the final printing of the paper in the TRANSACTIONS this point can be clarified for future use.

AUTHORS' CLOSURE (Mr. Wiesner): Yes, this was an error when the paper was originally prepared. What was labeled is not actually the *net latent heat*. The *net latent heat* must be taken only up to the saturation line, and the Figs. 5 and 6 as they appear in this volume of TRANSACTIONS have been changed so that the presentation here is academically correct. The quantity so labeled in the original illustrations, however, which allows for some pressure drop and some expansion in the inlet pipe (at constant temperature) is normally very nearly equal to the *net latent heat* for the refrigerant vapors under discussion here. This is because the constant temperature lines for these refrigerants are very nearly constant enthalpy lines in this region.

1672

No. 1672

FREE PISTON COMPRESSOR IN AN AIR-CONDITIONING SYSTEM†

By R. J. McCrory*, R. W. King**, and J. H. McNinch‡, Columbus, Ohio

THE FREE-PISTON refrigerant compressor had its origins in the continuing interest of the American gas industry in increasing the use of its product—gaseous fuels. Nearly 5 years ago, the American Gas Association posed this question to industry and technology: "How can the potential heat energy in natural gas be transformed into a refrigerating effect with high efficiency and low first equipment cost in such a way as to be attractive for residential air-conditioning service?" One of Battelle's recommended approaches to this problem was to combine 2 well-known and highly efficient systems, *viz:* the spark-ignition gas engine and the vapor-compression refrigeration cycle. The resulting combination would be compatible with the large number of existing air-conditioning systems employing commercial refrigerants, and could be installed as a remote unit for new or replacement systems. The combined efficiencies of the gas engine driving a vapor compressor indicated operating costs highly attractive to the home owner.

Although the conventional internal combustion gas engine has many advantages, its first cost, maintenance costs, and noise level might be deterrents to its general acceptance as an air-conditioning power source. However, the free-piston engine compressor uniquely combines the power generation and vapor compression functions in a single unit in a way which promises to overcome the primary objections cited.

Fig. 1 shows schematically the basic free-piston refrigerant compressor configuration as compared with an engine-driven and an electric-motor-driven refrigerant compressor. All 3 units can function in the same type of system using either Refrigerant 12 or 22 with an air-cooled condenser.

The predominant feature of the free-piston refrigerant compressor is its unusual simplicity. With the exception of pressure-actuated valves which have slight

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† This paper is based on a program of development sponsored at Battelle by the American Gas Association.

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** Assistant Chief, Mechanical Research Division, Battelle Memorial Institute.
‡ Project Leader, Mechanical Research Division, Battelle Memorial Institute.

Presented at the Annual Meeting of the American Society of Heating Refrigerating and Air-Conditioning Engineers, Lake Placid, N. Y., June 1959.

motion, the only moving part is the piston. Power developed in the combustion chamber is transmitted directly through the piston to the refrigerant. With few parts and essentially no critical fits, the production cost of the free-piston compressor promised to be low. The inherent simplicity of the free-piston unit would also help to satisfy the requirements for reliability and dependability—two primary factors in establishing maintenance costs. Furthermore, extremely conservative design practices could be and were employed as the development proceeded. With

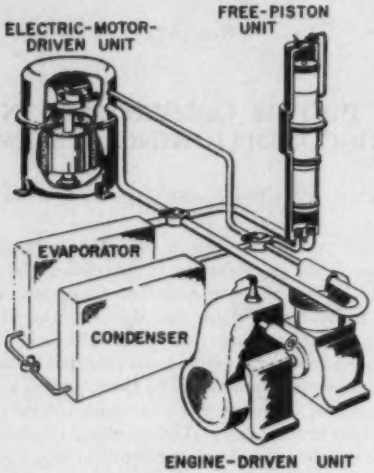


FIG. 1. . . .COMPARISON OF FREE-PISTON REFRIGERANT COMPRESSOR WITH CONVENTIONAL UNITS

essentially a single moving part, noise sources could be limited to the inlet and exhaust of the power end; pneumatic noises could be satisfactorily muffled.

DESCRIPTION

The free-piston refrigerant compressor has a 2-stroke-cycle combustion chamber on the top end and a compressor cylinder with conventional inlet and discharge valves on the bottom end. The diameters of both power and compressor cylinders are 2¾ in., and the nominal stroke of the piston is 3 in., resulting in nominal displacements of 17.8 cu in. The operating frequency is 1500 cpm with a condenser saturated temperature of 110 F. The refrigeration capacity under these conditions is 3 tons.

Fig. 2 is a photograph of the latest laboratory unit with which much of the present development work is being conducted. A prominent feature of this unit is the air-cooled combustion cylinder. At the present time, development of the free-piston

refrigerant compressor for use in a gas residential air-conditioning system has proceeded to a point where technical feasibility has been established. Many of the details have been worked out and several units are operating successfully in the Battelle laboratory under simulated field conditions.

Further development must be done to satisfy the requirements for residential air-conditioning service. Although the unit has been designed with life and durability as prime considerations, considerable work in the laboratory and in the field remains to satisfy the specified 2000-hr service life and 10,000-hr replacement life.

FIG. 2. . . .CURRENT LABORATORY UNIT

Further development will also improve its performance and decrease its operating cost.

The free-piston refrigerant compressor can be adapted for other applications and other fuels, including LPG, gasoline, and fuel oil, on residential heat pumps, automotive air conditioners, and truck and railroad car refrigerators. After specific development, the basic advantages of the free-piston compressor can be expected to provide attractive units for these applications.

GENERAL OPERATING CHARACTERISTICS

Fig. 3 schematically shows the general design of the compressor. Although this paper concentrates on the refrigerant compressor end of the cylinder, a brief summary of the operation of the entire unit is in order as general background. The combustion chamber employs loop scavenging with air inlet and exhaust ports in the cylinder walls. Air for scavenging and combustion enters the counter chamber through reed valves where it is compressed and delivered to the scavenge air box.

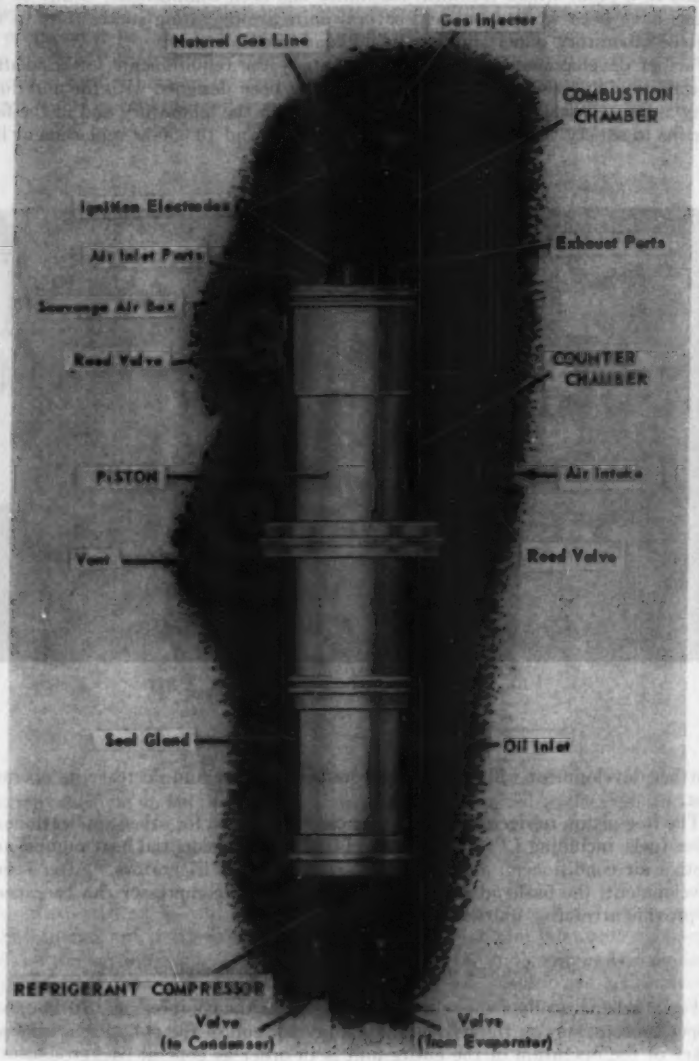


FIG. 3. . . .FREE-PISTON REFRIGERANT COMPRESSOR

When the piston uncovers the air inlet ports on its down stroke, the air scavenges the spent products of combustion from the previous cycle through the exhaust ports, and a new air charge is entrapped in the combustion chamber prior to the piston's compression stroke.

As the piston covers the exhaust ports, a charge of natural gas is injected from the counter chamber actuated gas injector and, as the compression stroke nears completion, a full charge of gas and air is intermixed in the combustion chamber. Igni-

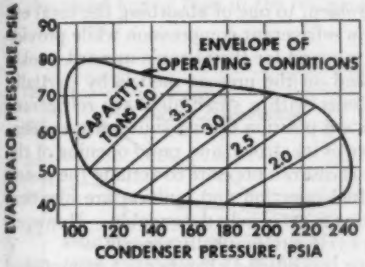
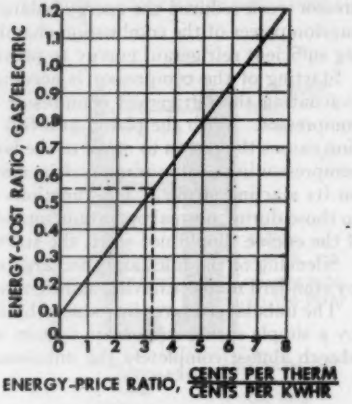


FIG. 4. . . .OPERATING CHARACTERISTICS OF FREE-PISTON REFRIGERANT COMPRESSOR



ASSUMED ENERGY REQUIREMENTS
Gas unit: 0.424 therm per hr
0.25 kw for cooling
Electric unit: 3.0 kw
Example: Columbus, Ohio
Gas: 6.55 cents per therm
Electric: 2 cents per Kwhr
Energy—price ratio: 3.27
Operating—cost ratio: 0.55

FIG. 6. . . .COMPARATIVE ENERGY COSTS OF 3-TON AIR-CONDITIONING UNITS GAS-FUELED FREE-PISTON UNIT *vs* ELECTRIC-MOTOR-DRIVEN UNIT

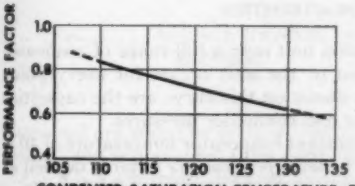


FIG. 5. . . .PERFORMANCE OF FREE-PISTON REFRIGERANT COMPRESSOR

tion occurs when a ground electrode on the piston crown passes in close proximity to a high-voltage electrode in the cylinder head, and a spark occurs.

As the combustion pressure drives the piston downward, compression begins in the refrigerant compressor and continues until the discharge valve opens and refrigerant flows to the condenser. When compression and displacement of the refrigerant dissipate all of the kinetic energy of the piston, the piston stops its downward travel with the compressor clearance volume filled with refrigerant at condenser pressure. The piston is started on its return by the re-expansion of the refrigerant in the clearance volume and, after inlet pressure is reached in the

refrigerant compressor, refrigerant at evaporator pressure provides the remaining energy for the upward stroke of the piston against engine compression pressure.

Operation of the free-piston compressor is obtained over a wide range of operating conditions by careful balancing of the compression and expansion energies in the combustion chamber and compressor. It was found that the refrigerant compressor automatically adjusts its compression ratio as the pressure ratio changes, so that the work of compression during any one cycle is constant in the range of required pressure ratios. This means that there is no need to modulate the amount of power generated in each cycle, and that a full charge of air and fuel can be used regardless of compressor conditions. This inherent characteristic of constant compressor work reduced the energy balance problem to one of absorbing the total expansion power of the combustion chamber in refrigerant compression while providing sufficient refrigerant energy to provide rebound for the piston's upward stroke.

Starting of the compressor is accomplished on the present models by partially evacuating the refrigerant compressor cylinder with a small auxiliary refrigerant compressor. With the piston at a full upward position during shutdown, evacuation causes the piston to move to the bottom of its stroke, and rapid opening of the compressor inlet valve admits refrigerant at shutdown pressure to actuate the piston on its starting stroke. The functions of fuel injection and ignition are identical to those during normal operation, and starting is extremely dependable. However, if the engine should not start, the starting cycle automatically repeats.

Silencing of the inlet and discharge to the combustion chamber is accomplished by standard mufflers having sufficient volume to satisfy the silencing requirements.

The unbalanced force introduced by the reciprocation of a single piston is isolated by a simple spring suspension system which permits the inertia of the cylinder to absorb almost completely the unbalanced forces.

PERFORMANCE CHARACTERISTICS

Fig. 4 shows the capacities of the free-piston unit over a full range of condenser and evaporator pressures. Stable operation of the unit occurs for every point within the bounds of the heavy line. Also shown on this curve are the capacities that are obtained for the various evaporator and condenser pressures.

Fig. 5 shows the *performance factor* for a constant evaporator temperature of 40 F over a range of condenser temperatures. The term *performance factor* is defined as the cooling effect divided by the heating value of the fuel delivered to the unit.

Fig. 6 is a nomograph from which the implications of this high performance factor as it relates to energy cost can be computed for specific natural gas and electric power rates. Although energy costs vary widely according to geographic location and level of utilization, the gas and electric rates for specific situations can be applied to Fig. 6 to obtain the relative energy costs. The example shown is for Columbus, Ohio, where for typical energy rates, the energy cost of the gas free-piston unit would be 55 percent of the energy cost of the electric air conditioner. Where the energy cost ratio is less than 6.5 (cents per therm/cents per kwhr), the energy costs of the gas free-piston air conditioner can be expected to be less than for the electric air conditioner.

Fig. 7 shows the noise spectrum of the present unit. The solid line is without shrouding of any kind, and the dotted line is the anticipated noise level with a cover of the type that would be used for a remote installation. Continuing work is further substantially reducing this noise level.

TECHNICAL DETAILS OF THE COMPRESSOR

This part of the paper considers several important details of the free-piston unit as a refrigerant compressor in a vapor-compression refrigeration system. Included is a discussion of design and development experience with the seal between the refrigerant compressor and the combustion system, the refrigerant compressor, and the starting system.

Refrigerant Seal: Sealing of the refrigerant within the compressor end of the unit is accomplished by maintaining an intermediate fluid in a gland around the compressor piston. By maintaining a sealing fluid such as refrigerant oil, at a pressure in the gland higher than in the refrigerant compressor, any leakage paths past the

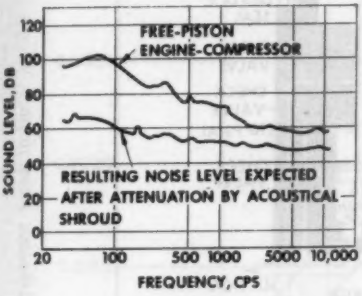


FIG. 7. . . . NOISE SPECTRUM OF THE FREE-PISTON COMPRESSOR

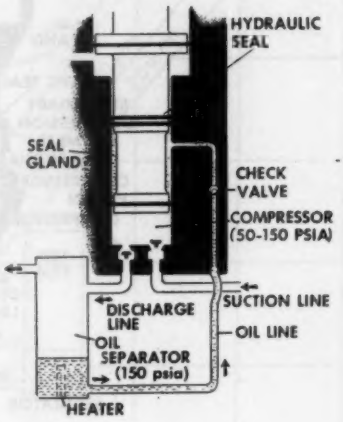


FIG. 8. . . . ORIGINAL REVERSE LEAKAGE SEAL

piston rod are sealed by oil flow from the gland into the compressor cylinder. The sealing fluid is prevented from escaping by hydraulic sealing rings.

Fig. 8 is a schematic drawing showing the original version of the reverse leakage seal as applied to the refrigerant compressor. The essential features of the system, aside from the sealing fluid gland, are:

(1) a system to recirculate automatically sealing oil that leaks from the gland into the refrigerant chambers;
(2) a system for maintaining a low concentration of refrigerant in the sealing oil; and
(3) a hydraulic seal for preventing excessive loss of sealing oil from the system.

In this original version of the reverse leakage seal, a circulation system was provided which started with oil in the separator at condenser pressure. The separator was maintained at an elevated temperature to strip refrigerant from the oil. A line from the bottom of the separator delivered oil to the sealing gland to maintain pressure in the gland above the compressor pressure during most of the cycle. Oil in the gland leaked through the clearance around the compressor piston into the compressor chamber. Discharge of the refrigerant from the compressor swept the reverse leakage oil through the discharge valve back into the separator.

An intensive investigation of the reverse leakage seal revealed 2 serious problems. First, the high shear area of the piston in contact with the seal resulted in viscous friction which was detrimental to high efficiency. The second, a much more serious problem, was the loss of refrigerant from the oil film on the cylinder wall which extended above the hydraulic sealing rings during the lower portion of the piston stroke. It was determined that the refrigerant, which was dissolved in the oil film during the upper part of the piston stroke, flashed rapidly from the film into the outside atmosphere. The obvious method of reducing the refrigerant loss was by

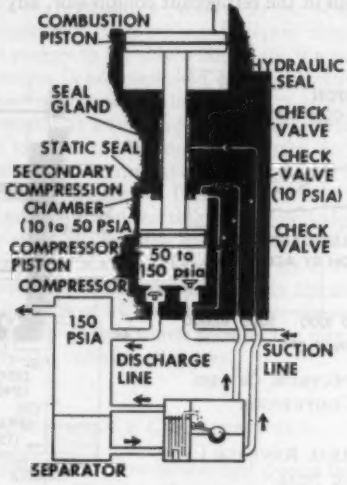
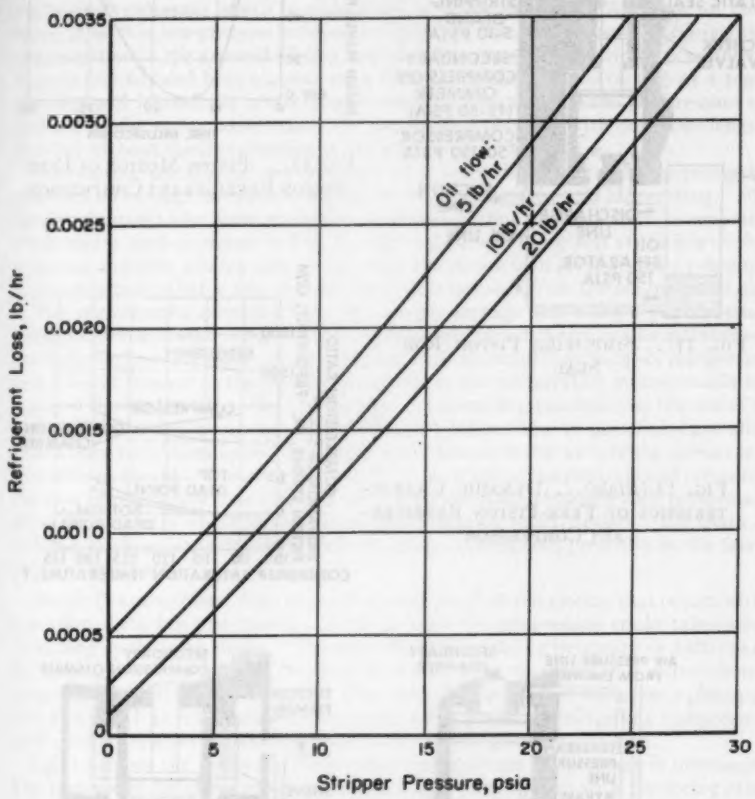


FIG. 9. . . .PISTON ROD CONFIGURATION OF REVERSE LEAKAGE SEAL

stripping the oil to a low refrigerant concentration by the use of high temperature and low pressure.

Fig. 9 shows a seal design devised to combine the objectives of low viscous friction, low wetted area on the piston skirt, and oil stripping. The principle of operation of the revised seal design was still the basic reverse leakage process. As shown in Fig. 9, however, a ¾-in.-diam piston rod which connected the combustion and compressor pistons decreased both the viscous shear area and the wetted area extending above the hydraulic ring. Furthermore, the volume above the compressor piston (termed the secondary compression chamber) provided a means for obtaining a low pressure for stripping the refrigerant from the sealing oil. As shown in Fig. 9, a check valve between the secondary chamber and the compressor inlet line caused the maximum pressure in the secondary to be compressor inlet pressure. This occurred at the top of the stroke of the piston, and the secondary pressure decreased as the piston moved down. A check valve between the secondary chamber and a stripper caused minimum secondary pressure to be applied above the oil level in a

low-pressure stripper stage. With this design, the oil passed from the separator into a first stripper stage where it was heated, still at condenser pressure. Expansion through a float valve provided a second stripping effect with the stripped refrigerant vented into the secondary compression chamber. The stripped sealing



¾ in. piston rod; Refrigerant-12; paraffin base oil; stripper temp, 250 F

FIG. 10. . . .PERFORMANCE OF PISTON ROD SEAL

oil then passed into the sealing gland where its pressure was sufficient to seal the refrigerant in the secondary compression chamber.

Fig. 10 shows the refrigerant loss that resulted with this seal configuration with a paraffin-base refrigerant oil, Refrigerant 12, and a stripping temperature of 250 F. As can be seen, low rates of refrigerant loss are obtained by expanding the secondary compression chamber down to low pressures.

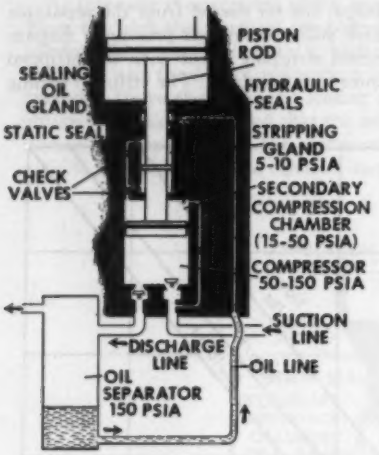


FIG. 11. . . .SIMPLIFIED PISTON ROD SEAL

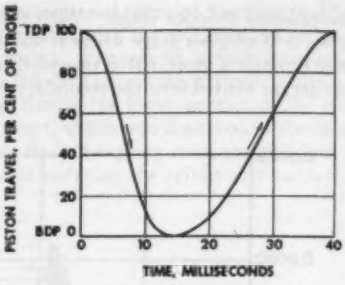


FIG. 12. . . .PISTON MOTION OF FREE-PISTON REFRIGERANT COMPRESSOR

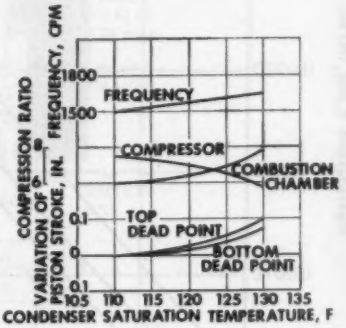


FIG. 13 (*Right*). . . .DYNAMIC CHARACTERISTICS OF FREE-PISTON REFRIGERANT COMPRESSOR

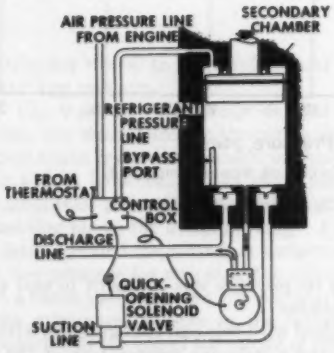


FIG. 14. . . .PUMP-DOWN STARTING SYSTEM

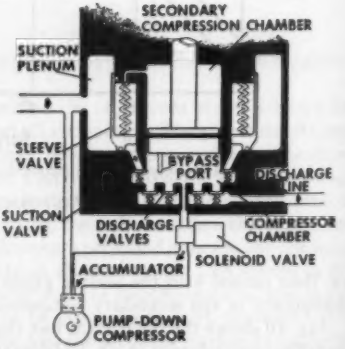


FIG. 15. . . .REDESIGN OF THE PUMP-DOWN STARTING SYSTEM

A large amount of operating experience was obtained with the seal design shown in Fig. 9. However, during the continuous simplification program, the external stripper and associated plumbing were eliminated by a new seal configuration.

Fig. 11 is a schematic drawing of this seal. As shown in the figure, the stroke length sealing oil gland has been replaced by a much shorter gland which is supplied with oil directly from the separator. Located below the oil gland is a second gland containing refrigerant which is maintained at a low pressure. With low oil flow rates, it is this low-pressure refrigerant chamber that determines the refrigerant concentration in the exposed oil film, and therefore, no external stripper is required. Viscous friction and heat transfer from the piston maintain the oil film at a temperature which results in a low concentration of refrigerant. The performance of this seal configuration is essentially the same as that obtained with the stroke length seal but without the complication of the external stripper.

Contracting metallic oil sealing rings have been developed which dependably maintain oil leakage rates below 0.05 lb per 1000 hr when the unit is operating. Although the rings also show promising shutdown sealing characteristics, a separate static seal is used as shown in Fig. 11, with a positive O-ring seal at the top of the stripping chamber coming into action when the piston is in its shut-down position. A piston-actuated valve also cuts off the system pressure from the sealing gland oil.

The performance obtained with the reverse leakage seal using a paraffin-base refrigerant oil as a sealing fluid indicates that a refrigerant charge can be maintained easily throughout a cooling season. In fact, with an initial overcharge of refrigerant and a liquid receiver in the system, no service to the refrigeration system would be required during the life of the compressor. A promising possibility is the use of a sealing fluid having refrigerant solubilities well below those in paraffinic-base oils. Fluids have been investigated in the laboratory having as low as 1/20 the refrigerant solubility as the oil. For any given condition of stripping temperature and pressure, the rate of refrigerant loss would be only 1/20 the value obtained with refrigerant oil, and the system would be, for all practical purposes, totally sealed. The characteristics of these low solubility fluids are being investigated presently in the free-piston compressor.

Design Considerations: Fig. 12 shows a schedule of piston motion that occurs with operation of the free-piston unit. As can be seen, the compression stroke takes only ½ as long as the inlet stroke. Therefore, for the operating frequency of 1500 cpm, the discharge valves must be designed as if they were operating in a conventional compressor operating at 2250 cpm. The design of the present compressor plenums and valving is an adaptation of a commercially available refrigerant compressor with modifications of the discharge valves to provide greater valve area.

Fig. 13 shows the limits of piston travel as condenser temperature is increased. The free-piston compressor is, in reality, a spring-mass system with the spring rate being determined by the combined pressure characteristics of the various chambers. An increase in the condenser pressure produces an increased spring rate of the system which results in a slightly higher operating speed. This increased speed is accompanied by a small shift in both the upper and lower dead points, the net effect being a negligible decrease in the stroke of the unit. An interesting characteristic of the free-piston compressor is its ability to ingest liquid refrigerant without ill effects. The liquid acts to decrease the compressor clearance volume which shortens the piston stroke.

Starting: Fig. 14 is a schematic drawing of the pump-down starting system used initially with the piston rod compressor design. The starting stroke of the free-

piston compressor is actuated by the refrigerant compressor cylinder. The piston is shown in its normal shut-down position which is at the upper limit of its travel. When the thermostat called for cooling, the pump-down compressor, a ⅓-hp unit, started and evacuated the compressor cylinder. When reduced pressures were reached, the piston was drawn down into the compressor cylinder. As the piston neared its lowest position, by-pass ports in the cylinder wall were uncovered which interconnected the compressor and secondary chambers. These ports were low enough that they were not uncovered during normal operation. The secondary chamber was then also evacuated to its normal running minimum pressure. When the control box received a signal that the secondary pressure was low enough to provide good starting dynamics, a quick-acting solenoid valve was opened in the compressor inlet line and the pump-down compressor was stopped. Refrigerant entering the compressor cylinder through the compressor inlet valve actuated the piston on its first upward stroke.

Three problems with the design shown in Fig. 14 led to a redesign of the starting system. First, location of a solenoid valve in the suction line with a large volume between it and the intake valve caused a slower than desired first piston stroke. Second, a properly sized solenoid valve was too costly for use in this system. Third, the pump-down process was made inoperable by a leaky compressor discharge valve.

Fig. 15 shows a redesign which replaced the solenoid valve with a sleeve valve to close off the compressor suction ports during shut-down. Secondary pressure was exerted on one side of the sleeve valve, and when the desired low secondary pressure was reached, differential pressure automatically snapped the starting valve open to admit the starting refrigerant charge to the compressor cylinder. The second design change was the insertion of an accumulator between the pump-down compresssor and the compressor cylinder. The accumulator, which was kept evacuated by the pump-down compressor, was opened to the compressor cylinder when the thermostatic signal opened the small solenoid valve. Evacuation of the compressor cylinder was so rapid that small leaks in the compressor discharge valve had negligible effect on the starting capabilities.

The record of starting reliability has been extremely good. On those occasions, however, when the unit does not start on the first cycle, a mis-start is sensed by the position of the starting valve which automatically closes after the mis-start, and the starting process automatically recycles.

At the present time, a mechanical starting system is being developed to eliminate the need for a pump-down compressor, a starting valve, and most of the control elements. This mechanical starting system is considered an approach to the ultimate in simplicity and reliability.

FREE-PISTON COMPRESSOR AS A PRODUCTION RESIDENTIAL AIR-CONDITIONING UNIT

Fig. 16 shows a complete condensing unit which represents the present developmental status of the free-piston compressor system. Individual components of the system are indicated, including a center of percussion suspension system for the compressor, exhaust and intake silencers, condenser and engine cooling blowers, oil supply reservoirs, controls, and the presently used pump-down starting compressor. The total size of this existing unit is 50 x 33 x 30 in.

During the rapidly advancing development program, further simplifications in this system can be expected. The pump-down compressor will be replaced by a mechanical system. The cooling of both the engine and condenser will be accom-

plished by one blower and the compressor sealant reservoir will be built into the condenser.

Fig. 17 is a partial adaptation of the free-piston compressor for low-cost production. The unit shown here has the sleeve-valve-type pump-down system. Further simplification and fewer parts would result with the mechanical starting system.

A thorough analysis has been made of the anticipated cost of manufacturing a gas free-piston air-conditioning system. The primary objective was to predict first cost relative to an equivalent electric air-conditioning system and, therefore, evaluate the economic feasibility of the free-piston compressor. Despite the operating

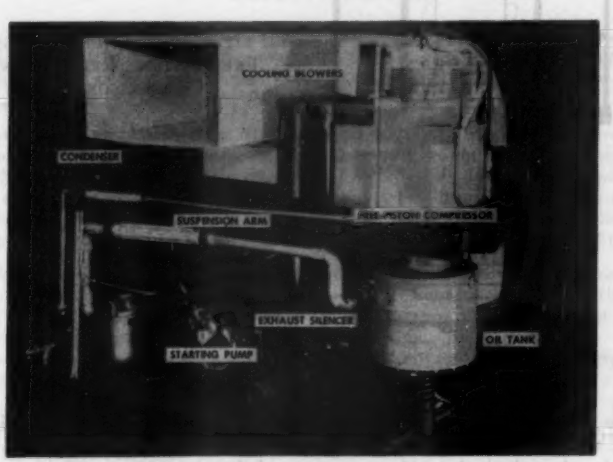


FIG. 16. . . .FREE-PISTON REFRIGERANT CONDENSING UNIT

cost advantage anticipated for the gas free-piston unit, marketing success requires a competitive first cost.

The electric and gas free-piston units which were compared were both of 3-ton capacity, had fan-cooled condensers, and were remote installations with the evaporators in the furnace. The free-piston unit was assumed to have a mechanical starting system, and the other improvements which reasonably can be expected to result from further development. For purposes of this analysis, costs were estimated at the system fabricator's level. Known prices of readily available components, such as condensers, fans, and electric compressors, were applied where possible. However, detailed costs of novel or unavailable components were carefully analyzed on a part-by-part basis. With an anticipated production rate of 15,000 free-piston compressors annually, the cost of the gas free-piston air-conditioning system should be only $15 more than the equivalent electric system. An increase in production to the range justifying higher tooling investments should more than make up the cost difference.

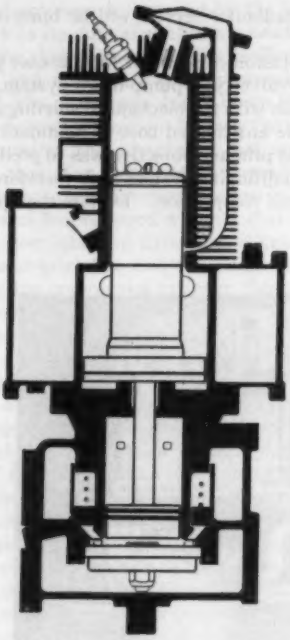


FIG. 17. . . .PRODUCTION DESIGN OF FREE-PISTON REFRIGERANT COMPRESSOR

Table 1 is a listing of the components required for both the gas free-piston and electric systems. The evaporator loop is not included. Cost estimates are presented for the principal novel components of the gas system. The other components of the gas free-piston system and all components of the electric system are

TABLE 1. . . .COMPARISON OF COMPONENTS REQUIRED FOR EACH SYSTEM

HERMETICALLY SEALED ELECTRIC COMPRESSOR	FREE-PISTON GAS-POWERED COMPRESSOR	
Electric compressor	Free-piston compressor	$76.77
¼-hp fan and motor	¼-hp fan and motor	
Fan capacitor	Fan capacitor	
Base and cabinet	Base and cabinet with acoustical treatment	
Control relay	Control relay	
Condenser coil	Condenser coil	
Electric contacter	Starting mechanism	$13.00
Capacitors, starting	Liquid receiver	
Capacitors, running	Engine oil tank	
Relay, starting	Ignition power supply	$15.00
Overload heaters	Exhaust muffler	
Dual pressure cutout	Air cleaner and silencer	

commercially available. Because there are variations among the costs for these components, however, they are not itemized in this paper but rather left to the experience of the reader.

The availability of an air-conditioning unit with a first cost equal to and an operating cost less than present electric units should have a major impact on the market. This paper is intended to show the progress toward this objective. When a manufacturer will be in a position to offer this unit remains to be seen. There can be little doubt, however, that the technical basis exists for successfully completing the development. Further, there is real promise that this will be accomplished soon.

DISCUSSION

M. B. GOETZ, Washington, D. C.: Much information was presented about the methods for keeping the refrigerant from escaping into the other cylinder. No mention was made of how the gases from the combustion are prevented from getting into the refrigerant. Can the authors comment on this?

AUTHORS' CLOSURE (Mr. McNinch): Air is quite insoluble in oil as compared to the refrigerant; therefore, the transfer by means of the oil film is not a serious problem. As to the exhaust products, they are discharged in the upper end of the unit—well away from the seal. The bounce chamber, which is beneath the large step in the piston, is a vented chamber which continually circulates fresh air. The main function of this chamber is to provide a vented area to separate the engine and compressor sections of the unit.

F. E. MEDON, Edgewater, N. J.: Are the data on the increase, with operating time, of the noncondensible gas in the refrigerant based on test results or on postulation?

Do the authors have any data on the increase in oil content in the refrigerant as compared with ordinary compression systems?

In a low-temperature application with a low-evaporator pressure, how would the piston be returned?

M. F. TOKACH, Milwaukee, Wis.: What average speed is obtained now? Is there any way of controlling the speed; that is, to lower it, or to modulate the speed with capacity or tonnage changes?

M. B. GOETZ, Washington, D. C.: The subject of low-temperature work has been brought up. Wouldn't it be true that to apply the same compressor to a low-temperature job the same results would be accomplished by using a higher-pressure refrigerant of the same design?

AUTHORS' CLOSURE (Mr. McCrory): Mr. Medon asked about the source of the data reported on noncondensible gas. Actual results were not obtained to get this information from the free-piston compressor itself. However, the compressor end was mocked up and we have a large number of hours on operation of the seal in both regards, namely, the escape of refrigerant from the system and the admitting of diluents from outside the system, and the large number of hours obtained with this setup indicated no problem with outside diluents.

Concerning the increase in oil content, quite a lot of work was done on the solubility of refrigerants in oil under various conditions, namely, temperature and pressure conditions which are critical to the stripping problem; but regarding their values as compared to conventional systems, there is no reason why they should be any different.

As to Mr. Medon's question about the return of the piston, it is correct that this has been strictly a design for air conditioning. There are 2 things which could be done to design the unit for low-suction head conditions. One is to provide a larger clearance volume on the compressor such that the piston would be returned primarily by re-expan-

sion rather than by inlet pressure. The other would be to make use of the pneumatic bounce chamber below the enlarged portion of the piston to return the piston on its compression stroke.

Mr. Tokach asked about piston speeds and the control of the unit. The mean piston velocity of the unit is 750 fpm which, as far as engine practice is concerned, is just above idle condition. The maximum mean piston velocity is 800 to 850 fpm—in that range. Peak piston velocity is approximately 30 fps.

With regard to the use of a modulated speed control for the unit, it could be done but it is doubtful whether it is worth it. One way would be to restrict the flow of gas to the unit to some extent. However, it is thought that the absence of controls such as we have with straight *on-off* operation is much to be preferred on a 3-ton unit at least. If this type compressor were to be adapted to a 50-, 100-, or 200-ton unit, which is technically feasible, the matter of speed control might be subject to wholly different considerations.

Concerning the subject of using a higher-pressure refrigerant as mentioned by Mr. Goetz, the answer to his direct question is "Yes." The important point to watch out for is the pressure ratios obtained. It is pressure ratios that are worked with more than absolute pressures, because pressure ratios represent energy and the attempt is to balance out energy. So, in going to some of these low-temperature applications with very high pressure-ratios, one would probably have to provide auxiliary air bounce in order to balance out the energies.

1673

No. 1673

AUTOMATIC COMPUTER FOR FAN TESTING

By C. H. POUNTNEY, JR.* AND D. W. SKIPWORTH**, CLEVELAND, OHIO

NORMALLY, fans are specified and purchased in accordance with standard fan test codes[1, 2] which have the approval of technical societies and fan manufacturers. These codes define properly fan characteristics and establish procedures for accurately determining fan performance, thus assuring accurate information for the design of equipment and air-handling systems.

The simplest method, which uses the conventional pitot tube, requires a minimum of 160 pressure and power readings, and thus draws heavily on the budgeted time of a development laboratory. In design studies, which are frequently of a cut-and-try nature, progress is limited severely. Furthermore, test time cannot be reduced simply by making fewer determinations, because of the resulting loss in accuracy. Other methods, involving nozzles or orifices, are therefore often used for fan appraisal in the laboratory and prove to be sound and efficient tools. This paper describes such a method which has been successfully extended to include rapid automatic computation and machine plotting of the fan performance curves with a high degree of accuracy.

Fig. 1 is a general view of the test chamber showing the blower mounting table and constant speed drive on the left and duct work running up to 2 exhaust blowers on the laboratory roof. Mechanical design is based on chamber specifications given in Plate XXI of Reference 3. All electrical and pressure measurements are ducted to the instrument console (shown in Fig. 2) from which speed, nozzle selection, and system resistances are controlled. Nozzles of 5-, 6- and 8-in. diam are used individually or in combination for flow measurements. When not in use the nozzles are sealed by inflating basketball type bladders in the nozzle throat. These bladders are mounted on air supply tubing and can be inserted or retracted from the nozzle, and inflated or deflated remotely from the console.

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** Senior Research Engineer, Viking Air Products Division, National-U. S. Radiator Corp.

[1] Exponent numerals refer to References.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N. Y., June 1959.

SPEED CONTROL AND MEASUREMENT

For test purposes, fans are mounted on the chamber as shown in Fig. 3. The mounting table is positioned by an electrical drive and the fan is connected with flexible couplings through the torquemeter (strain gage transducer) to the variable speed drive as shown. The shaft coupled to the blower on the right is connected to the tachometer pickup which is wired directly to the rpm indicator located at the instrument console (Fig. 2) giving continuous speed readings. In this way, speed may be maintained within ± 1 rpm of any desired setting. Since the tachometer is on the driven side of the torquemeter, the torque used to drive it is not measured. Where physical contact is not possible with the rotating blade, such as in axial fans,

FIG. 1. . . .FAN TEST CHAMBER WITH BLOWER MOUNTING TABLE AND CONSTANT SPEED DRIVE AT THE LEFT AND DUCT WORK RUNNING TO EXHAUST BLOWER ON THE RIGHT

speed is measured by interrupting a light beam to a photocell. This signal also appears on the speed indicator. Throughout all tests the speed is maintained constant to ± 1 rpm.

POWER CONSUMPTION MEASUREMENT

Shaft horsepower at a given speed is determined by measuring the torque input to the fan. This is done with the 0- to 200-in.-lb torquemeter shown mounted between the fan and the variable speed drive in Fig. 3. The output of the torquemeter in millivolts varies directly as the torque and thus at constant rpm the shaft horsepower does also. Since these tests are run at constant speed, the output of the torquemeter may be fed through a *weighting* network to one of the 2 pens of the function plotter to indicate shaft horsepower directly. This network is a simple voltage divider which is adjusted for different test speeds and desired horsepower ranges. The following illustrates the procedure to establish a 1.0 hp full-scale reading. At 1.0 shaft horsepower, Fig. 4 shows the torquemeter output to be 13.7 millivolts at 850 rpm. Since 5 millivolts will deflect the shaft horsepower recording

pen of the function plotter to full scale, the output of the torquemeter must be reduced to 5/13.7 to establish the 0 to 1.0 hp scale. In addition this signal may be further modified to correct the data to standard air conditions by multiplying this by the ratio of standard to ambient air density. This final ratio is then set up by adjusting the dial of the corresponding voltage divider.

In testing direct drive fans, watt input to the fan motor is measured with a thermal converter which produces a dc millivoltage proportional to watt input. This

FIG. 2. . . .INSTRUMENT AND CONTROL CONSOLE. THE 2-PEN FUNCTION PLOTTER IS IN THE UPPER LEFT OF THE CONSOLE

signal is also modified with the voltage divider network to obtain the desired span on the function plotter.

STATIC PRESSURE MEASUREMENT

A static pressure tap connects the test chamber with a manometer and a 0- to 0.2-psi differential strain gage transducer in the instrument console as shown diagrammatically in Fig. 5. Both zero and standardizing controls are provided for this measuring circuit making it possible to adjust the static pressure channel in the function plotter to any desired span. The unit is standardized by introducing a simulated pressure to the transducer and adjusting the pen to the proper setting.

In the present setup a calibrated resistance is applied to the static pressure transducer bridge to simulate 1.8 in. of water pressure differential. Thus, if it were desired that the full scale deflection (0 to 10) be equivalent to 2.5 in. of water, the pointer would be set, by adjustment to the bridge, at $1.8/2.5 \times 10 = 7.2$ on the scale. Data correction to standard air is made by multiplying the standard setting by the ratio of standard to ambient air densities.

MEASUREMENT OF AIR FLOW

Pressure differentials across the nozzles are also detectd with a 0- to 0.2-psi strain gage transducer. However, because flow through the nozzle is proportional to the

FIG. 3. . . .THE FAN MOUNTED ON THE CHAMBER, IS COUPLED THROUGH THE TORQUEMETER TO THE CONSTANT SPEED DRIVE. THE TACHOMETER IS ATTACHED ON THE DRIVEN SIDE OF THE SHAFT

square root of pressure differential, it is necessary to extract the square root of the signal given by the strain gage. This is accomplished electronically with a square root extracting circuit. In addition, because there are 7 possible nozzle combinations with the 5-, 6-, and 8-in. nozzles, it is necessary to further modify the output of the square root extractor to compensate for each combination. This is done by placing the output of the square-root-taking circuit across a voltage-dividing network which has successive steps proportional to each nozzle constant. Table 1 shows the manner in which this is arranged.

Circuit parameters, such as the transducer and potentiometer constants, are compensated for in the process of standardizing this circuit for the desired flow span. As in the static pressure circuit, a calibrated resistor placed across the transducer bridge simulates a differential pressure by which standardization of the selected nozzle combination can be achieved. Set up in this manner the signal, which now represents cfm at standard conditions, actuates the drive of the chart paper in the function plotter.

OPERATION OF THE COMPUTER TEST CHAMBER

Approximately 15 min is needed to mount a conventional fan on the test chamber. The fan mounting table is electrically controlled for rapid alignment with the variable speed drive and torquemeter. System resistance is controlled through a hydraulically actuated damper which modifies the flow of air from 2 exhaust blowers mounted on the laboratory roof at the end of the duct shown in Fig. 1. Nozzle selection made at the console is based on the size of unit being tested. Normally, one nozzle is sufficient to cover the entire normal operating span of a fan. If, however, accurate data at lower rates can be obtained only by changing nozzles during the test, this may be accomplished in approximately 2 min. Because the smallest

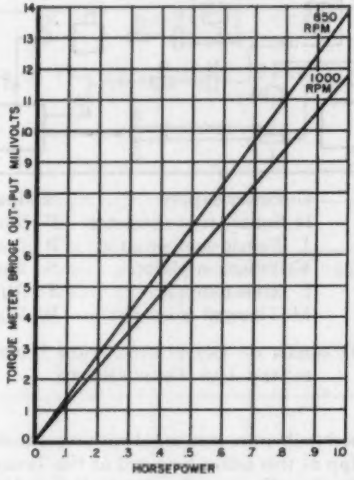


FIG. 4. . . .VARIATION IN TORQUEMETER BRIDGE MILLIVOLTAGE OUTPUT WITH SHAFT HORSEPOWER

nozzle presently used is 5 in. performance below 300 cfm cannot be obtained with sufficient accuracy except for static pressure at no delivery. Using all nozzles simultaneously a flow of 5000 cfm can be accurately measured.

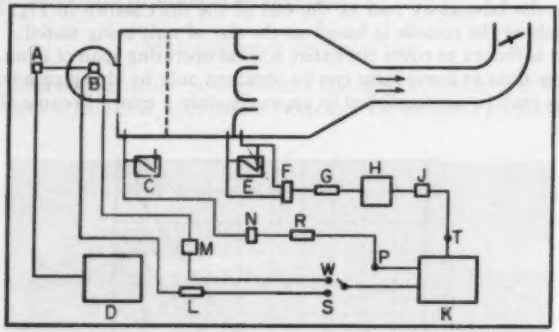
Complete test time is approximately 1 to 2 min at one constant speed. Similar determinations at other speeds can each be made in 1 to 2 additional min. Fan performance tables can thus quickly be determined without lengthy calculation, because it is less time consuming to run a test at various speeds than to compute performance.

EVALUATION OF PERFORMANCE

Fig. 6 shows the performance curve of a 10-in. forward-curved fan as traced out by the function plotter. The rapid fluctuation which appears throughout the major portion of the curve represents actual fluctuations of flow conditions encountered

throughout the range of performance. In transposing this curve to a master for reproduction only the mean value of amplitude is used.

The probable percentage of error introduced by the instrumentation is less than 2 percent. As a means, however, of verifying the accuracy of this equipment, re-



A Tachometer	G Power supply	N Transducer
B Torquemeter	H Square root extractor	P Static pressure signal
C Manometer	J Nozzle compensator	R Power Supply
D Speed indicator	K Function plotter	S Shaft horsepower signal
E Manometer	L Power supply	T Air flow signal
F Transducer	M Thermal converter	W Watts input signal

FIG. 5. . . .DIAGRAM OF INSTRUMENTATION USED IN COMPUTING FAN PERFORMANCE

sults obtained with this chamber are compared with results obtained with the same fan in Plate V[4] test setup at this laboratory and at the Texas Engineering Experiment Station, College Station, Texas. Results of all 3 of these determinations are given in Fig. 7 which shows extremely close agreement between the static pressure *vs* flow curves. Differences between the brake horsepower curves exist because measurements made in this equipment do not include the belt friction which is the case when calibrated motors are used to determine power consumption.

TABLE 1. . . .VALUES OF STEPPING RESISTORS IN NOZZLE SELECTOR CIRCUIT

NOZZLE COMBINATIONS DIAMETER, INCHES	RESISTANCE, OHMS
5	540.5
6	778.3
5 & 6	1318.8
8	1383.7
5 & 8	1924.2
6 & 8	2162.0
5, 6 & 8	2702.5

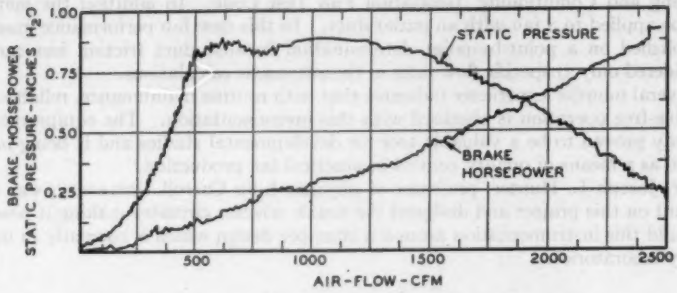


FIG. 6. . . .TRACE OF FAN PERFORMANCE CURVES OBTAINED WITH THE AUTOMATIC COMPUTER

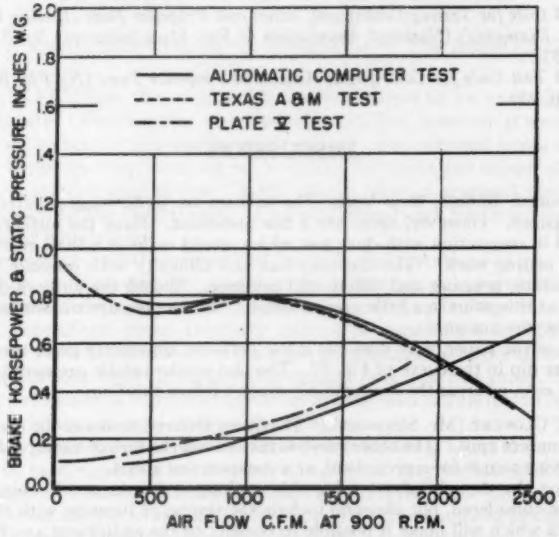


FIG. 7. . . .COMPARISON OF PERFORMANCE OF A SINGLE 10-IN. FAN OBTAINED WITH THE AUTOMATIC COMPUTER, A CONVENTIONAL PLATE V CHAMBER AND A PLATE V TEST CONDUCTED AT TEXAS ENGINEERING EXPERIMENT STATION, COLLEGE STATION, TEX.

For rating purposes, transducer readings are checked with manometers calibrated and positioned according to Plate XXI of Reference 3 and the new proposed Air Moving and Conditioning Association Fan Test Code. In addition the method can be applied to a fan with an outlet duct. In this case fan performance must be established on a point-by-point determination because duct friction loss can be considered only at specific flow rates in the automatic calculations.

Several months' experience indicates that with routine maintenance, reliable and trouble-free operation is obtained with this instrumentation. The equipment has already proved to be a valuable tool for developmental studies and is being introduced as a means of quality control in practical fan production.

Dr. Joseph L. Hunter, professor of physics, John Carroll University, was consultant on this project and designed the nozzle selector circuitry making it possible to build this instrumentation around a chamber design which is currently in use in many laboratories.

REFERENCES

[1] *Standard Test Code for Centrifugal, Axial and Propeller Fans* (AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS and the National Association of Fan Manufacturers, *NAFM Bulletin* No. 110, 1952).

[2] *ASME Test Code for Fans* (American Society of Mechanical Engineers, P.T.C. 11–1946).

[3] *Standard Code for Testing Centrifugal, Axial and Propeller Fans (Having Wheels Less Than 12-in. Diameters)* (National Association of Fan Manufacturers, *NAFM Bulletin*, No. 116, 1953).

[4] *Standard Test Code for Centrifugal, Axial and Propeller Fans* (*NAFM Bulletin* No. 110; *op cit.* p. 21).

DISCUSSION

J. B. GRAHAM, Buffalo, New York: The authors are to be congratulated on a very interesting paper. However, there are a few questions. Have the authors ever tried this method in connection with duct test which would make it a little more useful for general fan testing work? The discusser has had difficulty with pressure transducers for sensing static pressure and differential pressure. Would the authors discuss their installation at this point in a little greater detail? Is the pressure transducer something developed by the authors?

In Fig. 6 of the paper, why does the static pressure apparently go to zero at shutoff just after the dip in the curve of Fig. 6? The plot marked static pressure takes a drop and goes to zero, whereas the fan certainly doesn't follow this line.

AUTHORS' CLOSURE (Mr. Skipworth): The curve referred to shows the static pressure build up from zero rpm. The other curve is taken from the torque meter, a strain gauge, and varies with torque for a given load, at a constant test speed.

Concerning the thought about using this equipment for ductwork, this application has not been considered, but plans do include the testing of furnaces with this chamber by ductwork which will make it possible to connect to the equipment any furnace that is to be tested and it will be possible to test a furnace with normal operating conditions and the furnace itself acting as the system resistance. At this time, the authors see no reason why ductwork cannot be substituted for the furnace and measurements taken in that manner.

The pressure transducers are standard pressure transducers equipped for either measuring static pressure or differential pressure. The unit used has a zero to 0.20 pounds per sq in. range, and the power supply is designed for this pressure transducer.

1674

No. 1674

EVALUATION OF FILTERS FOR REMOVING IRRITANTS FROM POLLUTED AIR

By N. A. RICHARDSON* AND W. C. MIDDLETON**, LOS ANGELES, CALIF.

IT IS UNLIKELY that the *smog* problem will be *solved* on an area-wide basis in the near future. Pending this more desirable solution, however, it seems reasonable to attempt to control the quality of air brought into enclosed living and working spaces. Unfortunately, data are not available for use in the design of air filtering equipment for this application. With this in view, a research program was initiated to establish the effectiveness of various filter types for removing the irritants from smog polluted air.

Los Angeles *smog* currently is identified by such manifestations as eye, nose, and throat irritation, reduced visibility, and damage to agricultural crops and materials. The agents in polluted air which produce irritation and crop damage have not been identified. Filter effectiveness, therefore, cannot be established directly in terms of a change in chemical concentration, but must be measured in terms of the decrease in *smog* manifestations which attends a filter installation. In this study, differences in the sensory response of human subjects to filtered and unfiltered environments are used as a primary measure of filter effectiveness.

It should be recognized that such a method of evaluating filters does not insure the removal of carcinogens or other ingredients which may be harmful to health. However, it has not been established that the current level of community air pollution has a detrimental effect on health. When more is known about health aspects, it should be possible to design filtering equipment to remove causative agents.

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Presented at the meeting of the Chapters Regional Committee for Region 4 of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Los Angeles, Calif., May 6, 1957, and at the Annual Meeting of the AMERICAN SOCIETY OF HEATING REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N. Y., June 1959.

Improvement of human comfort and job performance efficiency is felt to be ample justification for the use of filters.

PROCEDURE

General: The tests were carried out in 2 rooms of the Pacific Telephone and Telegraph Building in downtown Los Angeles. One of the rooms was located on the 7th floor and the other was immediately above it on the 8th floor of this building.

FIG. 1. . . .ONE OF THE TEST ENCLOSURES SHOWING FILTER INSTALLATION

The rooms had identical floor plans, and were remodeled by the Telephone Company so that they were as identical as possible in all other respects.

In each test room there were approximately 20 young women who served as subjects during the testing program. The women in both rooms worked as typists and had similar levels of job responsibility.

Outside air was brought directly into the rooms in a single pass through the window-mounted ventilator-filter units shown in Fig. 1. Eight room-air changes per hour were provided regardless of the filter material being tested. Smoke tests indicated that the ventilator systems increased the pressure in the rooms to a level sufficient to prevent air from flowing into the rooms through the doors when they were opened; thus, the air brought into the test rooms was completely controlled.

It is important to note that these experiments were performed under completely real conditions. Neither the job situation nor the air pollution conditions were artificial.

Filter evaluation was accomplished by placing test filters in the ventilator units of one room while *dummy* filters matched for impedance to air flow were placed in

the ventilator units of the other room. The sensory response of the subjects to their immediate air environment was measured by means of a Daily Questionnaire, which was filled out by each subject. The Daily Questionnaire provided a composite measure of irritation which presumably indicated the subject's total response to the immediate atmospheric environment. The test filters were switched periodically so that the room occupants were alternately exposed to test and control conditions. Although the occupants knew the general nature of the test, they were never told whether the filters within their room were active or not.

The Daily Questionnaire was passed out each work day at 11:00 a.m. by the room supervisors, and concurrently, physical measurements of the air were made in each of the 2 rooms and immediately outside the building. Four different measures of the physical composition of the air were made. These included a measurement of particulate matter, nitrogen dioxide, and total oxidants by 2 different methods. Measurements of temperature, relative humidity, and room illumination were also obtained.

A preliminary sampling program was undertaken to establish the reliability and sensitivity of the questionnaire method of measuring sensory irritation, and to determine if differences existed in the response of the two groups. During this period *dummy* filters were placed in the ventilator units of both rooms. The results of this study were very encouraging. Good consistency was obtained between the response of individuals within a given test group as well as between the average response of the 2 test groups. The maximum possible response on the Daily Questionnaire is 22 irritation units. It was found from this study that a 3-unit variation of the Mean Index of Irritation of either test group is statistically significant to the 5 percent level.† The correlation between the mean response of the 2 groups during this period was 0.93.

Evaluation of Activated Carbon: Activated carbon was selected as the first filter material to be tested because some preliminary evidence indicated that this material was effective in reducing *smog* manifestations. This information, however, was inadequate, in terms of quantity and scientific control, to be of use in the design of activated carbon filters to meet specified requirements.

The filter unit in which the activated carbon was placed for testing is illustrated in Fig. 2. The filter bed, which has a thickness of 1⅛ in. and an effective area of approximately 10 sq ft, was filled with a 50-min‡ coconut shell carbon.

If the rates of adsorption of the irritating substances by the filter bed are controlled by the rate at which the materials diffuse into the pores of the activated carbon and not by the rate of diffusion from the air stream to the surface of the carbon (this is generally the case), then the mass transfer rate is essentially independent of air velocity in the filter bed. Under this condition, the time the air remains in contact with the filter bed becomes a significant parameter in predicting changes in concentration of the irritants. The effectiveness of activated carbon, therefore, was evaluated as a function of air detention time, t_d, where

$$t_d = 60lA_sP/Q \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

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† A three unit variation in the Mean Index of Irritation would occur less than 5 times in 100 by chance alone.

‡ Based upon the standard test for chloropicrin *breakthrough*.

and

t_d = air detention time within the filter bed, seconds.
l = actual bed thickness, feet.
A_e = effective area of filter bed, square feet.
Q = air flow rate, cubic feet per minute.
P = bed porosity, percent voids.

The time during which the air is detained in the filter bed has been calculated by using the rather standard assumption that the activated carbon bed has a

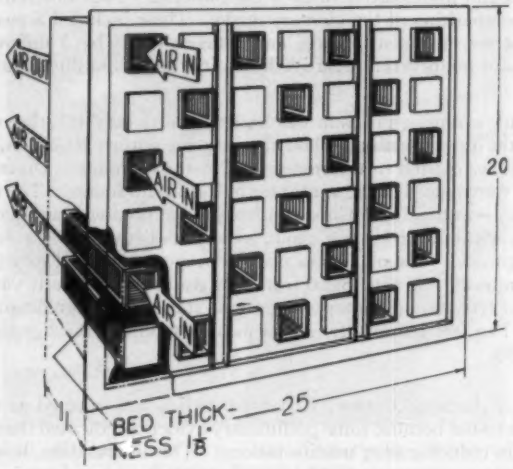


FIG. 2. . . .ACTIVATED CARBON AIR FILTER

porosity of 50 percent. Laboratory measurements indicate the actual porosity was approximately 47 percent.

For practical reasons, changes in air detention time were accomplished in 2 ways. The first reduction in air detention time was achieved by replacing one of the 2 activated carbon filter units, originally contained in each ventilator, by a *dummy* filter. Thus, air detention time was cut in half while the velocity of air past the activated carbon remained constant. Subsequent reductions in air detention time were achieved by blocking off portions of the remaining filter, so that bed thickness remained constant, but the air velocity was increased.

The first activated carbon tested gave off a definite odor shortly after it was put into operation. It was possible that this carbon had been used previously and not properly reactivated; therefore, it was replaced, after a period of 6 weeks, by new carbon. Although the new activated carbon also gave off a *sweet* odor after an operating period of several weeks, it was not changed again during the course of the experiment, in order that some indication of filter life might be obtained. It was originally planned to change the carbon whenever the effectiveness of the filter in

reducing sensory irritation decreased significantly. In this manner, it would be possible to determine the extent to which the reduced effectiveness was due to the accumulated exposure of the carbon to polluted air. However, a significant decrease in effectiveness did not occur during the estimated 650 hours of filter operation.

Evaluation of a Particulate Filter: Subsequent to the evaluation of activated carbon, a highly efficient particulate filter was tested using the same procedures. A filter carrying a guaranteed maximum penetration of less than 0.03 percent by particles 0.3 microns in diameter, based on particle count, was selected for this purpose. Because a significant portion of the particulate matter in the atmosphere on *smoggy* days is smaller than 0.3 microns, several samples of particulate material in the air stream before and after filtration were taken to determine if the filter would remove the smaller particles. The samples were collected using a thermal precipitator. The electron micrographs of Fig. 3 are presented to suggest the effectiveness of this filter in removing submicron particles. Based upon an actual count of particles having diameters greater than 0.01 microns, the filter was found to be 89 percent effective. It is interesting to note that the mean particle size in the sample taken before filtration was approximately 0.05 microns and after filtration was approximately 0.1 microns.

RESULTS

General: The time required for a particular filter test was necessarily determined by the frequency and intensity of *smog* conditions. The data presented were obtained during 123 sampling days from the 15th of May through the 15th of November, 1956.

Fig. 4 presents measurements of irritation, NO_2, and oxidant level, obtained simultaneously within the 2 test rooms and immediately outside the building, plotted as a function of sampling date. Certain effects of the different filters upon irritation scores and chemical measurements as well as some correlations are readily apparent in glancing at this figure. These effects and correlations are discussed in some detail later in this section.

An indication of error in the physical measurements was obtained by simultaneously sampling air at a single location with the 3 sampling units used during the filter evaluation program. Simultaneous measurements were taken on 12 different occasions under different *smog* conditions. The differences between the 3 sampling units were, on the average, 6 percent of the average measured value in the case of oxidant measurement by the phenolphthalein method, 8 percent of the average measured value in the case of oxidant measurement by the potassium iodide method, and 12 percent of the average measured value in measuring NO_2. Simultaneous measurements using the 3 particulate sampling units revealed an average difference of approximately 15 percent of the average measured value.

Correlations between measurements taken outside the building and in the non-filtered room, during the filter evaluation program, provide the best over-all indication of the reliability of the physical measurements. These correlations are influenced both by error of measurement and variable factors at each sampling location which influences the measurements in a differential manner. Because the active filters were periodically shifted from one room to the other and hence each room was in the non-filtered position for approximately half the duration of the study, the correlations apply equally well to both test rooms.

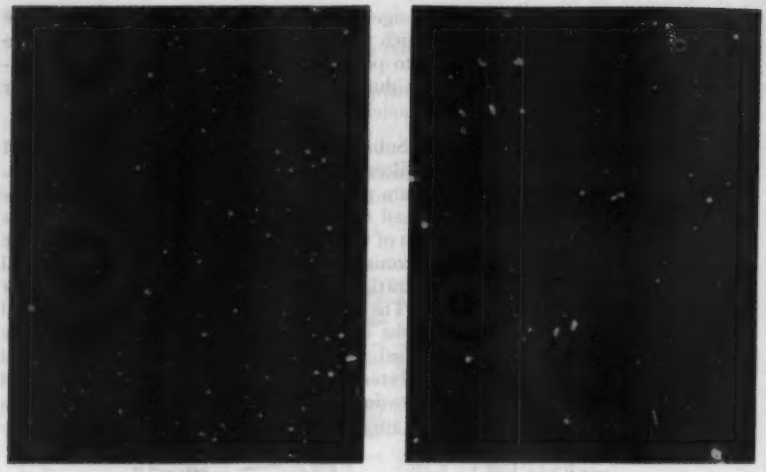


FIG. 3. . . .ELECTRON MICROGRAPHS OF PARTICULATE MATERIAL IN AIR ON A SMOGGY DAY

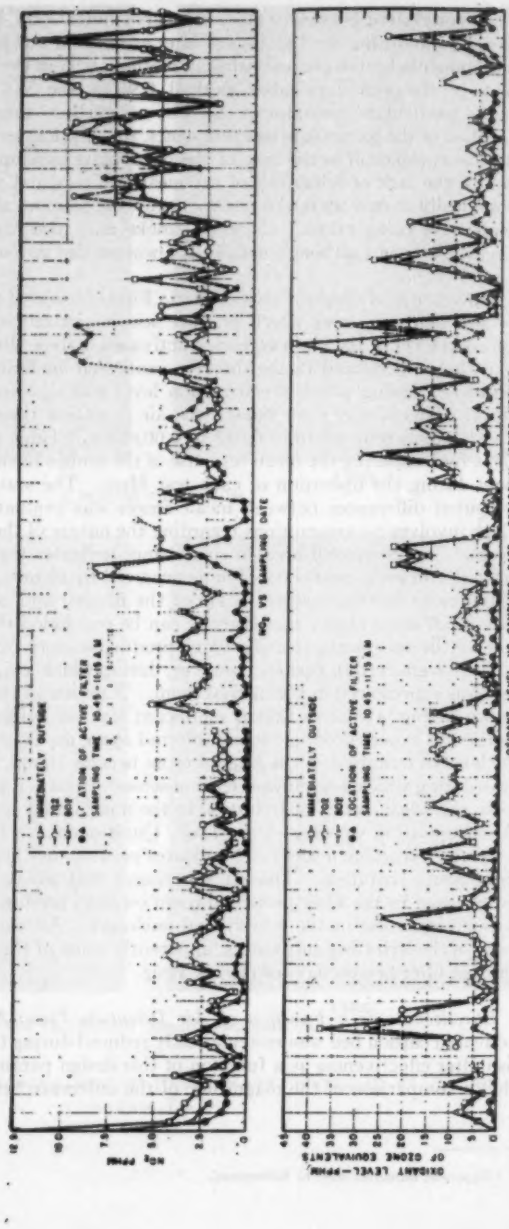
The correlation between values measured outside the building and in the non-filtered room during the 123 days of filter evaluation was 0.95 in the case of oxidant measurements by the phenolphthalein method, 0.93 in the case of oxidant measurements by the potassium iodide method, 0.90 for the NO_2 measurements, and 0.52 for the particulate measurements. The correlations thus indicate that, with the exception of the particulate measurements, the physical measurements were reliable. The low correlation in the case of the particulate measurements is apparently due both to the lack of reliability of the measurements and to the fact that dust was sporadically stirred-up in the test rooms by the room occupants while the measurements were being taken. Cigarette smoke may also have had some effect upon this measurement although usually the women did not smoke during office hours.

Evaluation of Activated Carbon Filters: First consider if activated carbon removes the substances in *smog* which produce sensory irritation. Fig. 4 shows that the irritation level in the room receiving activated carbon filtered air was always lower than that experienced in the directly ventilated enclosure if only those days are considered during which the irritation level was significant (3 or more irritation units). This is true even though the air detention time in the activated carbon filter bed was reduced from 0.032 to 0.0030 sec. Table 1 summarizes the data of Fig. 4 by comparing the mean response of the groups in the filtered and non-filtered room during the operation of each test filter. The statistical significance of the computed differences between mean scores was evaluated using the Sign Test[1] which involves no assumptions regarding the nature of the distribution of irritation scores. The computed level of significance indicates the probability that a given difference in mean scores could have occurred by chance. Because the differences in the mean irritation scores between the filtered and non-filtered test situations were in all cases highly significant it can be concluded that activated carbon does remove the substances in *smog* which produce sensory irritation.

There were certain periods, however, during which a significant amount of irritation was experienced in the filtered room. A review of the irritation measurements shown in Fig. 4 reveals that a significant level of irritation (3 irritation units or more) was experienced in the non-filtered space on 25 of 109 sampling days while the level of irritation in the filtered room became significant on 5 days. On the 5 days during which a significant level of sensory irritation was recorded in the filtered room the Mean Index of Irritation in the non-filtered space was 9 units or higher. Based upon the wording of the Daily Questionnaire a score of 3 indicates *barely noticeable* irritation, a score of 7 indicates *moderate* irritation, and a score of 11 indicates *severe* irritation. Thus it was found that *barely* noticeable irritation was experienced in the filter-protected room on days producing *high-moderate* to *severe* sensory irritation in the non-filtered enclosure. Although activated carbon does remove the irritating substances, apparently some of these materials did penetrate the test filter bed on days of intense *smog*.

Effectiveness as a Function of Air Detention Time: Air detention time in the activated carbon bed was systematically reduced during these experiments to establish filter effectiveness as a function of this design parameter. It should be noted that a comparison of the magnitude of the difference between means of the scores

[1] Exponent numerals refer to References.


IMMEDIATELY OUTSIDE
TOS
BOS
LOCATION OF ACTIVE FILTER
SAMPLING TIME 10:45-11:15 A.M.
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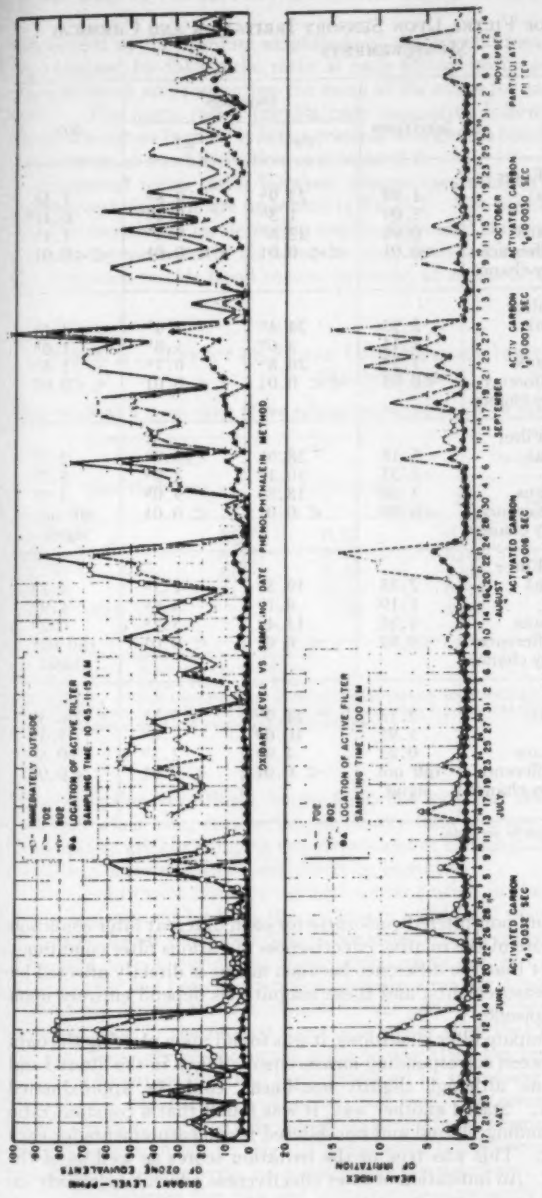

FIG. 4. . . .SENSORY IRRITATION, OXIDANT AND NITROGEN OXIDE CONCENTRATIONS VS SAMPLING DATE

TABLE 1....EFFECT OF FILTER UPON SENSORY IRRITATION AND CHEMICAL MEASUREMENTS

TEST CONDITION	IRRITATION	OXIDANTS		NO_2
		PH	KI	
[a] 0.032 Activated Carbon Filter				
Mean (non-filtered room)	1.99	23.9[b]	9.8[b]	1.5[b]
Mean (filtered room)	1.01	1.3[b]	0.49[b]	0.41[b]
Difference Between Means	0.98	22.6[b]	9.4[b]	1.1[b]
Probability that the Difference could have occurred by chance	<0.01	<< 0.01	<< 0.01	<<0.01
0.016 Activated Carbon Filter				
Mean (non-filtered room)	2.95	24.4[b]	8.4[b]	3.4[b]
Mean (filtered room)	1.41	3.6[b]	1.8[b]	1.6[b]
Difference Between Means	1.54	20.8[b]	6.7[b]	1.8[b]
Probability that the Difference could have occurred by chance	<0.01	<< 0.01	<< 0.01	<<0.01
0.0075 Activated Carbon Filter				
Mean (non-filtered room)	5.45	28.9[b]	13.9[b]	2.7[b]
Mean (filtered room)	2.35	10.1[b]	4 9[b]	5.7[b]
Difference Between Means	3.10	18.8[b]	9.0[b]	3.0[b]
Probability that the Difference could have occurred by chance	<0.05	< 0.01	< 0.01	diff not signif
0.0030 Activated Carbon Filter				
Mean (non-filtered room)	2.35	19.5[b]	7.3[b]	4.7[b]
Mean (filtered room)	1.19	8.1[b]	3.6[b]	4.9[b]
Difference Between Means	1.16	11.4[b]	3.7[b]	0.2[b]
Probability that the Difference could have occurred by chance	<0.01	< 0.01	< 0.01	diff not signif
Particulate Filter				
Mean (nonfiltered room)	2.13	14.9[b]	5.7[b]	6.3[b]
Mean (filtered room)	1.91	10.0[b]	3.4[b]	5.5[b]
Difference Between Means	0.22	4.9[b]	2.3[b]	0.8[b]
Probability that the Difference could have occurred by chance	diff not signif	< 0.01	< 0.01	<0.02

[a] Refers to Air Detention Time in seconds.
[b] Figures are in pphm.

recorded in the filtered and non-filtered enclosures for each different filter condition does not provide a measure of the relative effectiveness of various filter conditions. This results from the fact that the difference between means is directly affected by the magnitude of the measurements; and these magnitudes depend entirely upon the severity of a *smog* episode.

In seeking a way to compare filter conditions, it was found upon plotting the data that the relationship between corresponding measurements taken in the filtered and non-filtered test locations, although slightly non-linear, could be approximated closely by a straight line. Stated another way, it was found that a constant ratio existed between corresponding filtered and non-filtered test measurements for each different filter condition. This was true of the irritation scores as well as of the chemical measurements. An indication of filter effectiveness which is relatively in-

dependent of the absolute magnitude of the original measurements, therefore, can be obtained by taking the ratio of each filtered to its corresponding non-filtered measurement and computing the mean of the ratios for each of the test filter conditions. The mean ratios provide only suggestive information in the case of the irritation scores because it is not possible to state that an irritation score of 4 represents twice as much irritation as a score of 2.

Computed mean ratios between measurements taken in the filtered and non-filtered test situation are presented in Table 2. The mean ratios of irritation scores tend to increase as air detention time is decreased thus suggesting that filter effectiveness decreases as the air detention time in the activated carbon bed is decreased. An exception to this trend occurs, however, at an air detention time of 0.0030 sec.

TABLE 2. . . .EFFECT OF FILTER UPON SENSORY IRRITATION AND CHEMICAL MEASUREMENTS

(*expressed as a mean ratio of the measurements taken in the filtered and non-filtered rooms*)

TEST CONDITION		IRRITATION	OXIDANTS		NO_2
			PH	KI	
0.032[a]	ACF[b]	0.27	0.06[d]	0.04	0.26
0.016	ACF	0.32	0.16	0.23	0.61
0.0075	ACF	0.39	0.36	0.38	2.07
0.0030	ACF	0.34	0.43	0.54	1.23
Part. Filter[c]		1.06	0.67	0.64	0.86

[a] Refers to air detention time in seconds.
[b] ACF means: Activated Carbon Filter.
[c] Part. Filter means Particulate Filter.
[d] *Example:* During the operation of an activated carbon filter with an air detention time of 0.032 sec, the oxidant level in the filtered room was, on the average, 6 percent of the oxidant level in the non-filtered room, as measured by the phenolphthalein method.

This exception very likely results from the fact that these tests were run during a period when *smog* intensities were lower than average. Because the computed mean ratios are not completely independent of the magnitude of the original measurements, a lower mean ratio would be expected.

The mean ratios between oxidant concentration in the filtered and non-filtered areas as measured by both the potassium iodide and phenolphthalein methods increase as the air detention time is decreased. NO_2 concentration was reduced by activated carbon during the early period of its use, but after the same activated carbon filter had been in operation for approximately 2 months, the NO_2 concentration in the filtered room increased. Four periods occurred during which the NO_2 concentration in the filtered room was higher than that in the non-filtered room. Two of these periods occurred immediately after the only 2 occasions on which filter air detention time had been decreased by increasing the air velocity past the carbon. It has also been observed that high NO_2 concentrations in the filtered room generally occurred during periods of above normal temperature, although there were exceptions to this. The apparent breakthrough of NO_2 seems to have had no effect on the irritation scores in the filtered room. Likewise, no relationship was found between the NO_2 and irritation measurements in the non-filtered room.

Evaluation of a Particulate Filter: Inspection of the data presented in Fig. 4 and summarized in Table 1 clearly shows that the use of a highly efficient particulate filter to remove airborne particles has little effect upon the level of sensory irritation produced by *smog*. This suggests that particulate material is not important in the mechanism of sensory irritation. This conclusion is further substantiated by the fact that the computed correlation coefficient for the relationship between particulate count and sensory irritation, as measured in the non-filtered room, was very low (0.15), and the correlation between outside measurements of particulate count and oxidant level, where oxidant level has been shown to correlate highly with irritation, was 0.10. These results agree with evidence reported by the Stanford Research Institute to the effect that *removal of all of the particulate constituents had little effect on the eye-irritating action of artificial smog*[2].

Before dismissing particulate material as an important part of the Los Angeles problem, however, 2 points should be considered. First, although particulate material apparently is not directly related to the occurrence of sensory irritation, it is possible that particles are important when health aspects of the problem are considered. Second, although particles are not involved in the mechanism of eye irritation, they may play a part in the reactions by which the irritating substances are formed.

The particulate filter decreased the concentration of oxidants and nitrogen dioxide by a small amount as indicated by the data of Tables 1 and 2.

Correlations Between Measurements: Correlation coefficients have been computed between measurements taken in the room which was not filter protected to provide information which might be helpful in identifying the irritants in Los Angeles *smog*. These correlation coefficients are presented in Table 3. Based upon the standard error, correlation coefficients of 0.24 or greater are significant at the 1 percent level. The correlations between sensory irritation and oxidant concentration determined during this study are high and are of a magnitude similar to those obtained by other investigators[3]. Because errors of measurement will operate to decrease the correlation, the degree of relationship is probably higher than indicated. One cannot conclude from these correlations that the oxidants produce sensory irritation. It does appear, however, that the production of oxidizing material is linked in some way to the reactions which produce the irritating materials.

A plot of Mean Index of Irritation *vs* Oxidant Concentration reveals that the relationship is non-linear. The curvilinear correlation coefficients, η, which best describe the data, however, are very little different from the linear correlation coefficients presented. One reason for this apparently is the fact that η is based upon grouped data while the product-moment correlation coefficient is based upon ungrouped data, thus avoiding error from this source.

In looking at the data of Fig. 4, a complex relationship between oxidant and nitrogen dioxide concentrations seems to exist. The correlation coefficient obtained over the duration of this study, however, is very close to zero. The correlation between sensory irritation and nitrogen dioxide also is essentially zero.

The positive correlation between sensory irritation and temperature and the negative correlation between irritation and relative humidity might have been expected from results obtained by Roth & Swenson[4]. These investigators found increased irritation from formaldehyde vapors with high temperatures and low relative humidities. It also is interesting to note that a relationship exists between temperature and oxidant concentration. Therefore, the positive correlation be-

tween sensory irritation and temperature may result from increased sensitivity on the part of the test subjects at higher temperatures as well as possibly higher concentrations of the irritating substances.

The correlation between sensory irritation and particle count is not significantly different from zero. One may question the significance of this correlation in view of the previously discussed low correlation between the particulate measurements made outside the building and in the non-filtered room. A separate check on this information is provided in the correlation between outside measurements of particulate count and oxidant concentration, where oxidant concentration has been

TABLE 3. . . .PEARSON PRODUCT-MOMENT CORRELATION COEFFICIENTS COMPUTED FROM MEASUREMENTS TAKEN IN THE NON-FILTERED ROOM

Irritation	*vs*	Oxidants (Phenolphthalein Method)	0.81
Irritation	*vs*	Oxidants (KI Method)	0.81
Irritation	*vs*	NO_2	0.05
Irritation	*vs*	Particulate	0.15
Irritation	*vs*	Temperature	0.49
Irritation	*vs*	Relative Humidity	−0.24
Temperature	*vs*	Relative Humidity	−0.38
Temperature	*vs*	Oxidants (KI Method)	0.29
Oxidants (Phenolphthalein Method)	*vs*	Oxidants (KI Method)	0.88
NO_2	*vs*	Oxidants (KI Method)	−0.15

shown to be highly related to sensory irritation. This correlation is again not significantly different from zero.

It should be noted that the correlation coefficients presented were computed from measurements taken once a day at 11:00 a.m. It is possible that the correlations would be significantly different if based upon measurements made at other times of day. For example, it has been reported that sensory irritation persists long after the oxidant concentration has dropped[5]. Therefore, correlations based upon continuous measurements are important if one wishes to attempt the identification of the substances which produce sensory irritation.

CONCLUSIONS

1. A statistically significant reduction in sensory irritation is accomplished when Los Angeles *smog* is passed through activated carbon filters having air detention times at least as short as 0.0030 sec.

2. Although activated carbon removes *smog* irritants, some of the irritating materials apparently did penetrate the test filter bed on days of intense *smog*.

Note: The most demanding requirement of a filter system, designed to remove the irritants from area-wide pollution, is the purification of air which is drawn entirely from outside of the building. A considerable amount of experience has been gained, however, in the use of activated carbon for the purification of recirculated air[6]. This procedure reduces the amount of outside air which must be filtered, and also reduces the air-conditioning load in buildings where air conditioning is employed.

3. The differences in effectiveness of activated carbon filters with respect to variations in air detention time between 0.032 sec and 0.0030 sec are not statistically significant. However, the ratios between corresponding irritation levels in filtered and non-filtered atmospheres indicate a trend of decreasing effectiveness as air detention time in the filter bed is reduced.

4. The effectiveness of activated carbon in removing oxidants is a function of air detention time. Effectiveness will be improved very little by increasing detention time beyond 0.032 sec, and it falls off at an increasing rate as detention time is decreased below this point.

5. NO_2 is reduced by activated carbon during its early use, but after a period of time, the NO_2 concentration in the filtered atmosphere increases and is frequently higher than in the non-filtered atmosphere. The reason for this apparent breakthrough has not been studied.

6. The removal of airborne particles down to a diameter less than 0.05 microns does not significantly decrease sensory irritation resulting from Los Angeles *smog*. These results, together with the correlation data, show that particulate material is not directly related to the occurrence of sensory irritation.

7. Correlations computed from daily measurements obtained in a non-filtered atmosphere at 11:00 a.m. indicate that sensory irritation is highly related to oxidant level and moderately related to temperature. A slight negative correlation has been computed between irritation and relative humidity. The correlations between irritation and NO_2 and between irritation and particulate count are not significantly different from zero.

ACKNOWLEDGMENTS

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APPENDIX

PHYSICAL MEASUREMENTS

Oxidant Measurements: One measure of oxidant concentration was obtained using the phenolphthalein method, as described by Haagen-Smit and Fox.[7] Air was bubbled through the reagent for 30 min at the rate of 1 liter per minute. The solution was then analyzed using a spectrophotometer at a wave length of 550 millimicrons.

The oxidant level was established in terms of ozone equivalents by calibrating the solutions against known quantities of ozone from a high-voltage discharge ozonizer. The ozonizer was calibrated by passing its output through potassium iodide and titrating with sodium thiosulfate.

Another measure of oxidant level was obtained by bubbling air, at the rate of one liter per minute, for 30 min, through potassium iodide in a phosphate buffer at a Ph of 7 to 7.2. The solution was then analyzed on the spectrophotometer at a wave length of 352 millimicrons.

Nitrogen Dioxide Measurement: Nitrogen dioxide was measured using a solution of sulfuric acid, N-(1-naphthyl)-ethylenediamine dihydrochloride and acetic acid, according to the procedure suggested by B. E. Saltzman.[8] Air is bubbled through the solution for 30 min at the rate of 0.8 liters per minute. The solution is analyzed on the spectrophotometer at a wave length of 550 millimicrons.

Particulate Count: Particulate material was collected from the air with continuous thermal precipitators. The details of sampling equipment and counting techniques, using the electron microscope, are discussed in Reference 9.

DISCUSSION

G. B. PRIESTER, Baltimore, Md.: The expression of *50-minute carbon* was used. Could the author elaborate on this? Also could he clarify how different air detention times were obtained?

C. R. HIERS, New York, N. Y.: Can the author comment as to how the dust samples were taken to permit the use of electron microscope technique in obtaining the micrographs of the airborne particles?

J. E. HAINES, Minneapolis, Minn.: Why was all outside air used when this is not the normal approach used in air conditioning? Was a prefilter used ahead of the activated charcoal filter?

G. L. BIEHN, Staunton, Va.: Was any form of electrostatic filter used in the work?

AUTHORS' CLOSURE (Mr. Richardson): Activated carbon is rated using a standard test developed by the U.S. Army Chemical Warfare Service. The test entails passing chloropicrin vapor at a definite concentration and flow rate through a fixed weight of carbon and recording the length of time before the vapor is detectable at the outlet of the activated carbon bed.

Regarding the second question of Mr. Priester, air detention time in the filter bed was changed in two ways: (a) air detention was changed from 0.032 sec to 0.016 sec by decreasing the bed thickness at constant air velocity, and (b) air detention time was changed from 0.016 sec to 0.0030 sec by using a bed of constant thickness and changing

the air velocity through the bed. In that the mass transfer process involved is not controlled by the diffusion rate from the air stream to the surface of the carbon but rather by the rate at which the adsorbed material diffuses into the pores of the carbon, the mass transfer rate becomes essentially independent of velocity. This makes total bed contact time the important parameter and justifies decreasing air detention time by increasing the air velocity in the bed.

Concerning the method of taking dust samples, the details of this process are described in Reference 9 of the paper. A thermal precipitator was used to collect the sample upon a microscope slide. In the thermal precipitator the sampled air is drawn through a narrow slit where it is subject to a temperature gradient. The particles contained in the air stream are deposited with very high efficiency as a consequence of this temperature gradient. The exposed slide is then shadow cast and photographed under an electron microscope. Particle counting and sizing is done by eye from a negative print.

As regards Mr. Haines' question about the use of outside air, the objective was not to establish an economical air filtering installation but was rather to test an activated carbon bed for its effectiveness in alleviating eye irritation. Using 100 percent outside air was the most severe test condition and therefore was the condition selected. A course glass fiber prefilter was used to protect the carbon bed.

An electrostatic precipitator was not tested, however it is believed that the results of such a test would be similar to those obtained with the high efficiency particulate filter. It will be recalled that with the very efficient removal of particulate material we did not obtain a significant decrease in sensory irritation.

1675

No. 1675

FUNDAMENTAL ANALYSIS OF CHIMNEY PERFORMANCE†

By W. G. COLBORNE* AND W. C. MOFFATT**, KINGSTON, ONT., CANADA

A GREAT part of the investigations into chimney performance have been done on masonry chimneys of certain fixed sizes. While the results of this work give the draft and efficiency to be expected from a chimney of the same size and construction as the test chimney, practically no attempt has been made to generalize chimney design using a fundamental approach. The purpose of this paper is to present results of the initial work toward a fundamental treatment of chimney performance. Using the theory as developed, a further paper on design will be prepared which will make possible the sizing of a chimney to meet any given requirements.

Masonry chimneys were not used in this work due to high costs of construction and difficulty in changing diameters and heights. Instead, it was decided to use aluminum tubes of 4- and 6-in. diam from which chimneys of various heights could be easily fabricated. Having checked the fundamental theory, calculations could then be made which would apply to chimneys of any material and of any dimensions.

BASIC THEORY

Since the gas within a chimney is at a higher temperature than the surrounding air, it is less dense and therefore a pressure difference exists across the chimney wall at any given point. It is this difference in pressure, or draft, which causes the

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† This paper is based on the M.Sc. thesis of W. C. Moffatt in the Department of Mechanical Engineering of Queen's University.

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upward flow of the heated gas in the chimney. In an ideal chimney (one in which there is no friction and no heat loss) the draft at the thimble wall will be $(\rho_0 - \rho_1)gH$. In actual practice such a condition could not exist and any calculation of actual or available draft must include an accounting of the friction and heat losses.

The loss of draft due to friction stems from two sources, (1) viscous and dynamic losses along the length of the chimney and (2) dynamic losses created by the chimney entrance. The general equation for friction loss resulting from the flow of fluids in pipes is

$$h_f = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right)$$

In this expression f is a friction factor whose value depends on the relative roughness and diameter of the tube and the temperature, velocity and viscosity of the gas flowing. In the laminar region (Reynold's Number less than 2300) the value of f is given by

$$f = \frac{64}{N_{Re}}$$

For flows involving Reynold's Numbers in excess of 2300, the value of f may be found from Moody's friction chart[1] which illustrates graphically the relationship between f, N_{Re} and e/d.

Obviously the determination of an overall friction factor K for the chimney is complicated by the chimney entrance, so that the value of f determined from Moody's chart must be augmented by an additional factor to account for the dynamic losses occurring in this region. Piggott[2] found that the total loss occurring in an elbow or tee could be assumed to be made up of 2 elements, in addition to ordinary pipe friction: (1) a minimum true bend loss for smooth pipe, depending only on r/d where r is the bend radius and d the pipe diameter; and (2) a turbulence loss due to roughness affected by the values of N_{Re} and e/d. The total friction loss for a chimney expressed in terms of velocity heads would therefore be given by an expression

$$K = M\left(\frac{r}{d}\right)^x + N(f)^z + f\left(\frac{L}{D}\right)$$

where

M, N, x and z = constants to be determined experimentally.

L = total length of chimney and thimble under consideration.

In the aluminum chimneys, as in most actual chimneys, the entrance to the vertical flue is sharp-edged and therefore the term r/d may be neglected. Thus the friction loss in the chimney may be assumed to be a function of friction alone and will take the form

$$K = N(f)^z + f\left(\frac{L}{D}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The second important factor which reduces efficiency is heat loss. As the gas flows up the chimney, it loses heat to the chimney surface which in turn is cooled by the surrounding air. This results in a reduced mean chimney temperature and con-

[1] Exponent numerals refer to References.

NOMENCLATURE

Symbol — Quantity	Dimensions
C_p = specific heat at constant pressure.	$H/M\theta$
D = draft.	M/LT^2
d = diameter.	L
G = mass flow rate per unit area.	M/TL^2
g = acceleration due to gravity.	L/T^2
H = height (physical).	L
h = heat transfer coefficient.	$H/TL^2\theta$
h_i = inside surface film coefficient.	$H/TL^2\theta$
h_f = fluid friction loss in pipe.	L
k = thermal conductivity.	$H/LT\theta$
L = total length (physical).	L
P = pressure (absolute).	M/LT^2
r = radius of pipe elbow.	L
ρ, $\Delta\rho$ = density, density difference.	M/L^3
T = temperature.	θ
T_m = temperature of chimney gas in an isothermal flow situation which would give the same chimney performance as in the actual case where the gas enters at T_1 and leaves at T_2.	θ
U = overall heat transfer coefficient.	$H/TL^2\theta$
μ = absolute viscosity.	M/LT
V = velocity.	L/T
ν = kinematic viscosity.	L^2/T
x, z = experimental exponent constants.	0
Z = height (physical) above datum.	L
M, N = experimental constant factors.	0

Dimensionless Groups

K = Euler's number = isothermal friction factor.

f = Darcy's friction factor = $\dfrac{2gdh_f}{LV^2}$.

N_{Re} = Reynold's Number = dV/ν.

e/d = relative roughness of pipe.

Subscripts

1 = chimney inlet.
2 = chimney outlet.
m = mean.
o = ambient.
S = surface.

Dimensions

The M L T θ H system of dimensions is used in this paper. The symbols and units are as follows:

M = mass in pounds.
L = length in feet.
T = time in seconds.
θ = temperature, Fahrenheit degrees.
H = heat in Btu.

sequently a lower available draft. The rate of heat transfer from the hot gases to the chimney wall is dependent on the temperature difference between the gas and the wall, the area of chimney surface in contact with the gas and the inside film coefficient. The following expression[3] has been found to yield acceptable values for the film coefficient:

$$\frac{h_i d}{k} = 0.023 \left(\frac{dG}{\mu}\right)^{0.8} \left(\frac{C_p \mu}{k}\right)^{0.4}$$

where

h_i = inside film coefficient.
d = chimney diameter.
k = thermal conductivity of chimney gas.
G = mass flow of gas per unit area.
μ = absolute viscosity of chimney gas.
C_p = specific heat of chimney gas at constant pressure.

For the aluminum chimneys used in the tests outlined in this work, the thermal resistance of the walls could be neglected, and the exterior surface, being reasonably

shiny, could be assumed to be non-radiating. The only other factor affecting the overall heat transfer coefficient U in addition to h_i would therefore be the outside film coefficient h_0. It has been found[3] that for vertical heated flues, h_0 may be given by

$$h_o = 0.29 \left(\frac{T_s - T_o}{d} \right)^{0.25}$$

where

T_s and T_o = chimney surface and ambient temperatures respectively.

Thus it is possible, by integrating over the entire length of the chimney, to determine the heat loss from the chimney and from this, the chimney exit temperature. For actual chimneys, the calculation of U would have to be modified to account for the thermal resistance of the wall.

Appendix A outlines the development of the basic equation describing the performance of a chimney. This expression is

$$\frac{D}{\Delta\rho gH} = \frac{\Delta\rho_m}{\Delta\rho} \left(1 - \frac{K\rho_1 V_1^2}{2\Delta\rho gH} \right) \quad \ldots\ldots\ldots \quad (2)$$

where

$D/\Delta\rho gH$ = an efficiency term relating actual draft D to ideal draft.
$\Delta\rho_m/\Delta\rho$ = a measure of efficiency drop due to heat loss.
$K\rho_1(V_1)^2/(2\Delta\rho gH)$ = the pressure drop due to friction in the chimney with isothermal flow.

A plot of $D/\Delta\rho gH$ *vs* $K\rho_1 V_1^2/2\Delta\rho gH$ would therefore yield a straight line for isothermal flow*, the loss in efficiency due to friction being given by the vertical distance between the line $D/\Delta\rho gH = 1.0$ and the isothermal line. Further, the drop in efficiency due to heat loss is given by the distance between the actual efficiency curve and the isothermal line.

Tests were carried out on the aluminum chimneys with a view to checking the theoretical equation, and noting the effects of a number of variables on chimney performance. It was considered that if these effects could be explained by the terms of the efficiency equation, then this equation could be assumed to apply for all chimneys, provided suitable accounting was made of the altered roughness and heat transfer characteristics of the chimney walls.

One variable, infiltration, while not a factor in an ideal laboratory chimney, was considered to be of sufficient significance in actual chimneys to warrant tests to study its effect. No provision was made for the effects of leakage in the fundamental Equation 2 but this aspect of chimney performance is of practical interest and is treated in Appendix B.

DESCRIPTION OF APPARATUS

Fig. 1 shows the layout of test and measuring equipment. The air measuring station was located in a 1.50-in. ID copper tube, with 42 diameters of tube length ahead of the measuring station to ensure fully developed velocity profiles. The air quantity was determined in the usual manner by running a 20-point pitot tube

* Such a plot shown as Fig. 4.

traverse at various flow rates, from which a calibration curve was plotted. Velocity pressures were measured on a micromanometer which could be read to 0.001-in. of water.

The air was heated in an electric furnace having accurately controllable electric heaters to maintain the air entering the chimney at the desired temperature. The

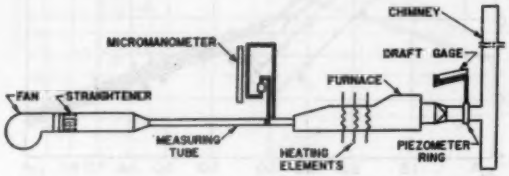

FIG. 1. . . .LAYOUT OF CHIMNEY TEST EQUIPMENT

heated air passed from the furnace into the test chimney. The draft was measured at the piezometer ring as shown in Fig. 1; the draft gage was calibrated in position and had an accuracy of 0.001-in. of water.

Temperature measurements were made using copper-constantan thermocouples located at the chimney inlet and chimney outlet. Room temperature was measured on a shielded mercury thermometer, barometric pressure was measured using a

TABLE 1. . . .DIMENSIONS OF ALUMINUM CHIMNEYS TESTED

DIMENSIONS	4-IN. CHIMNEY					6-IN. CHIMNEY	
Inside diameter, inches	3.90					5.86	
Piezometer ring to chimney centre-line, feet	1.00					1.50	
Base of chimney to thimble centre-line, feet	1.00					1.50	
Piezometer ring to transition exit, inches	8.0					12.0	
Tube wall thickness, inches	0.056					0.068	
Height from thimble centre-line, feet	3.90	4.87	5.85	6.83	7.80	5.86	7.32
Height to diameter ratio = H/d	12	15	18	21	24	12	15

mercury barometer, and wet-bulb temperatures were found in the usual manner. Table 1 gives the dimensions of the aluminum chimneys used.

TEST RESULTS

K in the fundamental Equation 2 is a friction factor based on isothermal flow, defined as the head loss due to friction divided by the velocity head and depends only on N_{Re} for any given chimney. For the chimneys used in this work, K was

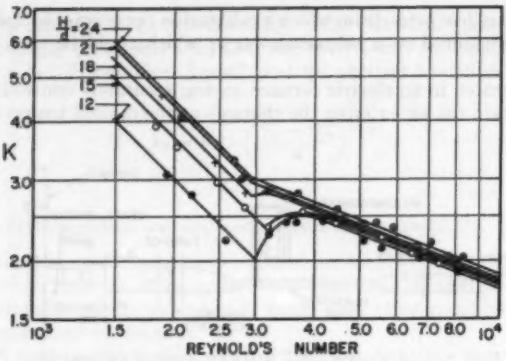

FIG. 2. . . .ISOTHERMAL FRICTION FACTOR K FOR 4-IN. DIAMETER ALUMINUM CHIMNEYS

determined experimentally with air flowing through the chimney isothermally at room temperature. The results of these tests for a 4- and a 6-in. diam chimney are shown graphically in Figs. 2 and 3 respectively.

The chimneys were then installed as shown in Fig. 1 and tested, with Reynold's Numbers at the piezometer ring ranging from approximately 1500 to 10,000. The velocity at the chimney inlet was varied from approximately 1.3 to 8.0 fps and the inlet air temperatures ranged from 230 F to 450 F. Fig. 4 shows a typical performance curve of a chimney using the coordinate system as just outlined. The curves shown in Fig. 4 are for a 6-in. diam chimney with $H/d = 15$ and no insulation. Fig 5 shows the same set of performance curves for a 4-in. diam chimney.

To illustrate the effect of varying H/d ratios, both the 6- and 4-in. chimneys were tested at various chimney heights. Fig. 6 shows the result of varying H/d for a 4-in. chimney with the inlet air density ratios $\rho_1/\Delta\rho$ maintained at 1.51 (this density ratio corresponds to an inlet temperature of approximately 420 F).

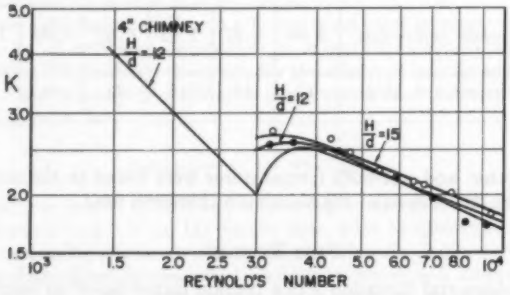

FIG. 3. . . .ISOTHERMAL FRICTION FACTOR K FOR 6-IN. DIAMETER ALUMINUM CHIMNEYS

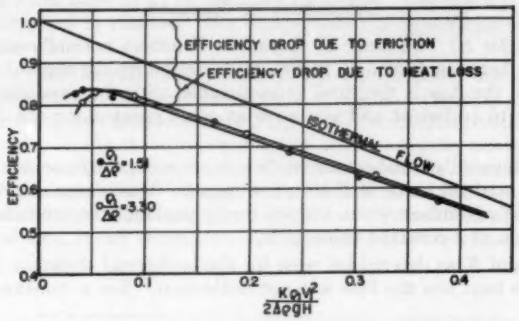

FIG. 4. . . . EFFICIENCY OF AN UNINSULATED 6-IN. DIAMETER ALUMINUM CHIMNEY WITH 2 DIFFERENT INLET TEMPERATURES (H/d = 15)

While the aluminum chimneys used were essentially non-radiating, the conduction and convection heat transfer was relatively high. Realizing that a masonry chimney would have a lower heat loss than the aluminum chimneys used, it was decided to test both 6-in. chimneys when covered with a 1-in. layer of 85 percent magnesia insulation. The results of these tests are shown in Fig. 7, again for the 6-in. chimney with H/d = 15. The curves for the uninsulated chimney are shown as well to illustrate the difference in performance. Since the isothermal line represents the limit of efficiency attainable, it was decided that tests using a greater thickness of insulation would yield little additional information.

Some additional tests with air leakage into the chimney were performed, the results of which are outlined in Appendix B. These tests were not extensive, rather they were brief runs to evaluate qualitatively the effect of air leakage on efficiency.

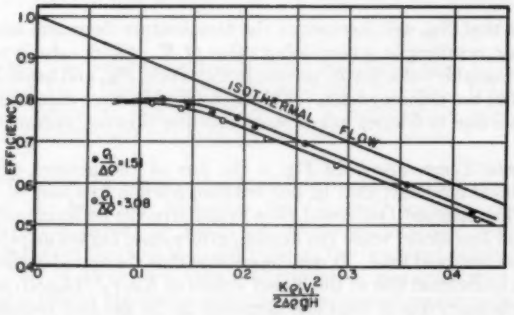

FIG. 5. . . . EFFICIENCY OF AN UNINSULATED 4-IN. DIAMETER ALUMINUM CHIMNEY WITH 2 DIFFERENT INLET TEMPERATURES (H/d = 15)

DISCUSSION OF RESULTS

Friction Factor K: The range of Reynold's Numbers normally encountered in chimneys lies between 1500 and 10,000, thus straddling the zone of critical flow. The nature of the flow is therefore unpredictable, with the possibility of changes from laminar to turbulent and *vice versa* at some point along the length of the chimney.

Since the Reynold's Numbers are in the critical zone, the losses are both viscous and dynamic, and the value of K therefore varies. This is particularly true at the lower Reynold's Numbers where viscous forces predominate, rendering impossible the assumption of a constant value of K.

The values of K as determined were for the isothermal situation; in the actual chimney with heat loss the flow was not isothermal. For a constant weight flow

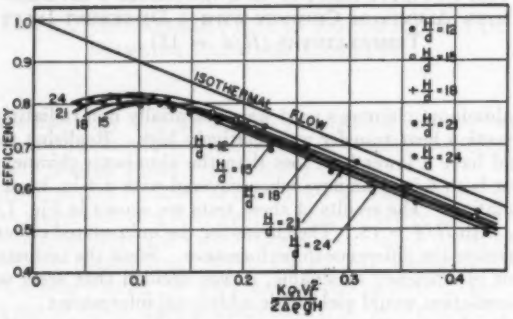

FIG. 6. . . . EFFICIENCY OF UNINSULATED 4-IN. DIAMETER ALUMINUM CHIMNEYS OF VARIOUS HEIGHTS (INLET TEMPERATURE = 420 F)

it can be seen that N_{Re} will increase as the temperature decreases along the length of the chimney, resulting in a decreasing value of K. Obviously, it would be awkward to use a variable value for K, so that in this work, N_{Re} and hence K, were based on conditions at the chimney inlet. This is justified in most cases since the greater part of the loss due to friction takes place near the chimney entrance.

Effect of Inlet Temperature: In Fig. 4 the line of 100 percent efficiency would apply to an ideal chimney, that is, one with no friction loss and no heat transfer. The straight line marked *Isothermal Flow* would give the performance of a chimney having friction loss alone, while the actual performance curves as plotted take care of both friction and heat loss. It may be noticed that the actual performance curves approach the isothermal line at the higher values of $K\rho_1 V_1^2/2\Delta\rho gH$, indicating that the drop in efficiency due to heat loss decreases as the gas flow increases. The loss in efficiency due to friction, however, increases at such a rate as to cause an overall drop in efficiency with increasing flow. The changing of the density ratio $\rho_1/\Delta\rho$

from 1.5 to 3.3 has little effect on chimney efficiency. The density ratios of 1.51 and 3.30 correspond to chimney inlet temperatures of approximately 420 F and 240 F respectively.

Fig. 5 shows that the efficiencies found for the 4-in. chimney are approximately the same as those found for the 6-in. chimney of Fig. 4. The only significant difference between the results shown in the 2 figures is the relative position of the density ratio curves. In Fig. 4 the curve of $\rho_1/\Delta\rho = 3.3$ is slightly above the curve of $\rho_1/\Delta\rho = 1.51$ whereas in Fig. 5 the opposite is true. Normally, it would be assumed that at a given value of $K\rho_1 V_1^2/2\Delta\rho gH$ the efficiency would increase as inlet temperature decreased due to a reduction in heat loss. On analysis of the effects of T_1, T_m, $\Delta\rho$ and $\Delta\rho_m$, it can be shown from Equation 2 that the change in efficiency

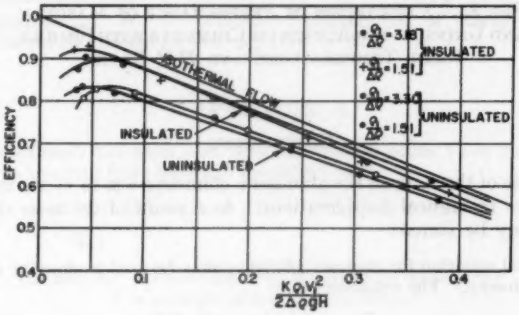

FIG. 7....EFFECT OF INSULATION ON THE EFFICIENCY OF A 6-IN. DIAMETER ALUMINUM CHIMNEY ($H/d = 15$)

is dependent on the manner in which T_m varies with respect to T_1. Under certain conditions of T_1 and T_m it is possible that the efficiency will increase with an increase in T_1.

Effect of Height: The height-to-diameter ratio was varied from 12 to 24 and once again as seen from Fig. 6 there was little effect on the efficiency. It is interesting to note that in this figure (as well as in Figs. 4, 5, 7 and 8) the maximum efficiency occurred at a $K\rho_1 V_1^2/2\Delta\rho gH$ of approximately 0.1 in all cases.

Effect of Insulation: Obviously the application of insulation to the chimney would decrease the heat loss and raise the efficiency for any given value of $K\rho_1 V_1^2/2\Delta\rho gH$. The extent to which the efficiency increased was of interest, however, and it was to this end that the tests on the insulated chimneys were performed. The 1-in. of 85 percent magnesia insulation was arbitrarily chosen, and would represent a greater insulation value than the normal masonry chimney. The efficiency of a masonry chimney of similar dimensions would therefore be expected to fall between the 2 curves of Fig. 7 at any given value of $K\rho_1 V_1^2/2\Delta\rho gH$.

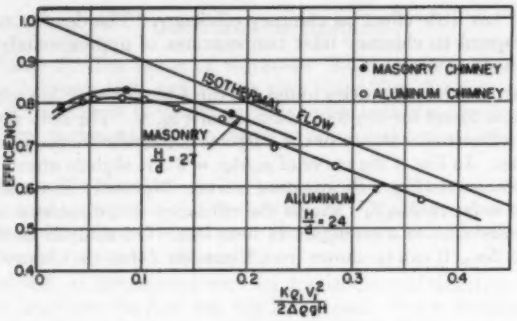

FIG. 8. . . . COMPARISON OF PERFORMANCE OF MASONRY AND UNINSULATED ALUMINUM CHIMNEYS WITH SIMILAR INLET TEMPERATURES AND H/d RATIOS

CONCLUSIONS

Main purpose of the tests on the aluminum chimneys was to verify the equations arrived at from theoretical considerations. As a result of the tests the following conclusions may be stated:

1. The general equation for chimney efficiency may be used to describe the performance of any chimney. The equation is:

$$\frac{D}{\Delta\rho g H} = \frac{\Delta\rho_m}{\Delta\rho}\left(1 - \frac{K\rho_1 V_1^2}{2\Delta\rho g H}\right)$$

2. The friction drop in a chimney may be calculated provided the nature of the flow and the surface roughness inside the chimney are known. From Figs. 2 and 3 the following empirical equations may be found:

$$K = f\left(\frac{L}{d}\right) + 52 f^{1.63} \text{ for turbulent flow } N_{Re} > 3100$$

$$K = f\left[\left(\frac{L}{d}\right) + 25\left(\frac{H}{d}\right)^{0.5}\right] \text{ for laminar flow } N_{Re} < 3100$$

The K as defined by these equations includes the loss at the tee connection at the chimney entrance as well as the loss in the chimney itself.

3. The effect of temperature on chimney performance is not readily predictable. Generally, for chimneys with large H/d, increasing the center-line chimney inlet temperature will increase the efficiency.

4. Increasing the H/d of a chimney will normally reduce the efficiency, other conditions being equal. However, the magnitude of the efficiency drop will depend to a large extent on the heat transfer characteristics of the chimney wall.

5. In order to show how the test results taken from a real chimney fit on the coordinate system used in this paper, Fig. 8 was plotted. The data of Achenbach and Cole[4] for a 7-in. ID brick chimney, 15.5 ft high were plotted on the same graph as the test results

from the 4-in. diam aluminum chimney with $H/d = 24$. The inlet temperature to the aluminum chimney was approximately 420 F, while the inlet temperature to the brick chimney was 600 F. Since the brick chimney had a slightly higher insulation value, it would be expected that the efficiency would be slightly above that for an aluminum chimney. The curves show the excellent correlation which was obtained.

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APPENDIX A

If it is first assumed that there is no heat loss between points 1 and 2 of the chimney (Fig. A-1), it is apparent from Bernoulli's equation that

$$SP_1 + VP_1 + Z_1 = SP_2 + VP_2 + Z_2 + \text{losses}$$

where

SP and VP = static and velocity pressures respectively.
Z = a height above a given datum.

Now $(Z_2 - Z_1) = H$ and since the flow is isothermal, $VP_1 = VP_2$.

Therefore,

$$h_1 = h_2 + H + \text{friction losses} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-1)}$$

where

h_1 and h_2 = static pressures in feet of chimney gas at points 1 and 2, respectively.

Expressing the friction losses as $KV_1^2/2g$, and converting to pressure units, Equation A-1 becomes

$$(h_1 - h_2)\rho_1 g = \left(H + \frac{KV_1^2}{2g}\right)\rho_1 g = P_1 - P_2 \quad . \quad . \quad . \quad . \quad . \quad \text{(A-2)}$$

Also, it is evident that

$$P_o = P_2 + \rho_o g H \text{ or } P_o - P_2 = {}_{,o}gH$$

where

P_o = ambient air pressure at the thimble level.

Therefore,

$$P_o - P_1 = (\rho_o - \rho_1)gH - \frac{K\rho_1 V_1^2}{2\Delta\rho g H} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-3)}$$

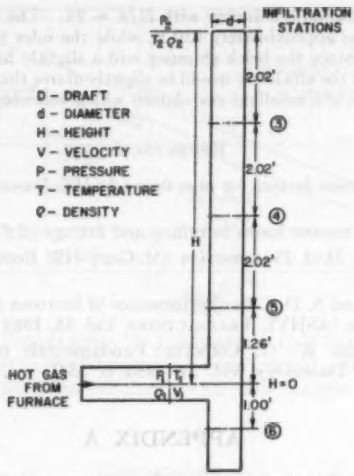

FIG. A-1. . . .SCHEMATIC DIAGRAM OF A TYPICAL ALUMINUM CHIMNEY SHOWING LOCATION OF INFILTRATION STATIONS (EACH STATION CONSISTS OF SIX ½-IN. DIAMETER HOLES EQUALLY SPACED AROUND THE CHIMNEY)

Now $P_o - P_1$ represents the pressure difference between the ambient air at the thimble level and the pressure within the thimble at point 1. Hence it is the draft.

Letting $P_o - P_1 = D$, and $\rho_o - \rho_1 = \Delta\rho$, and dividing by $\Delta\rho gH$, Equation A-3 becomes

$$\frac{D}{\Delta\rho gH} = 1 - \frac{K\rho_1 V_1^2}{2\Delta\rho gH} \quad \ldots\ldots\ldots \quad \text{(A-4)}$$

The term $D/\Delta\rho gH$ is an efficiency term, being the ratio of actual draft to ideal draft, and $K\rho_1 V_1^2/2\Delta\rho gH$ is a measure of the efficiency loss due to friction.

In order to modify Equation A-4 to account for heat loss, it must first be assumed that

$$T_m = T_1 - F(T_1 - T_2),$$

where

F is a constant. T_m (with its corresponding ρ_m) is the temperature of the gas in an isothermal flow situation which would give the same chimney performance as the actual case where the gas enters at temperature T_1 and leaves at temperature T_2. For the purposes of the calculations carried out in conjunction with this work, F was assumed (see Reference 5) to be 0.65. This value for F was confirmed during the experimental portion of the present work and from a paper dealing with tests performed on actual chimneys (see Reference 4). It is felt that this value ($F = 0.65$) may be used with

accuracy for any chimney; however, for chimneys above 30 ft in height the value remains to be confirmed.

If one defines $\Delta\rho_m = \rho_o - \rho_m$, it is apparent from the definition of T_m that for non-isothermal flow in a chimney

$$\frac{D}{\Delta\rho_m g H} = 1 - \frac{K\rho_1 V_1^2}{2\Delta\rho g H}$$

or

$$\frac{D}{\Delta\rho g H} = \frac{\Delta\rho_m}{\Delta\rho}\left(1 - \frac{K\rho_1 V_1^2}{2\Delta\rho g H}\right)$$

APPENDIX B

A short series of tests was performed on the 6-in. chimney, $H/d = 15$, to determine qualitatively the effects of infiltration air. For the purposes of these tests, 4 infiltration stations were incorporated (stations 3, 4, 5 and 6 in Fig. A-1), each consisting of six -½ in.

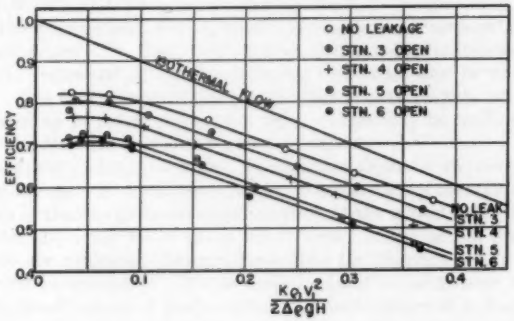

FIG. B-1. . . . EFFICIENCY OF UNINSULATED 6-IN. DIAMETER ALUMINUM CHIMNEY WITH INFILTRATION AT VARIOUS HEIGHTS (T_1 = 420 F, H/d = 15)

diam holes equally spaced around the chimney's circumference. The chimney inlet air temperature was maintained at approximately 420 F and draft and temperature measurements were made as for the normal draft tests.

The results of the tests are shown in Fig. B-1. It is immediately apparent from the curves that the lower the point at which the leakage air was introduced, the greater was the resultant drop in efficiency. This effect is readily predictable and may be attributed to 2 factors, *viz:*

1. The lower the infiltration station, the greater will be the draft at that point in the chimney, and consequently, the weight of infiltration air flowing through the holes will be greater and the resultant dilution of hot chimney gases with atmospheric air will be increased;

2. Cold air introduced at the lower stations will tend to give a lower temperature throughout the entire height of the chimney, with a greater reduction in efficiency than will the introduction of cold air near the chimney exit, which will affect but a short portion of flue.

DISCUSSION

This paper and the following paper, Chapter 1676, were discussed at the same time. See the Discussion beginning on p. 439.

1676

No. 1676

A NEW METHOD OF CHIMNEY DESIGN AND PERFORMANCE EVALUATION†

By W. C. MOFFATT*, AND W. G. COLBORNE**, KINGSTON, ONT., CANADA

THIS PAPER PRESENTS a practical design method for the sizing of chimneys based on the theories and tests outlined previously in "Fundamental Analysis of Chimney Performance."[1] The theory is general and is not limited to any specific size of chimney; industrial or small residential chimneys may be treated in the same manner. The design graphs apply to chimneys with heights up to 80 ft and flue gas flow rates up to 600 lb per hr, but these values may be easily extended by the theories presented here and in the previous paper.

The design method yields, first of all, the chimney diameter required based on a given set of conditions. This diameter results in a chimney of maximum efficiency. The second step in the design is the determination of the actual draft which will be obtainable under the same set of given conditions. If the actual draft found is not sufficient for the equipment being served, then the chimney height or the inlet temperature must be increased. If the draft is found to be greater than that required then the usual means of draft control, *e.g.* installation of a draft regulator, may be used.

The ratio of actual draft to theoretical draft is the efficiency, the actual being less than the theoretical by the 2 factors of friction and heat loss. If both these losses are accurately evaluated then the actual draft may be determined. In general, this is the procedure followed.

The design method presented here is based on steady state conditions and no account is taken of the effects of infiltration and wind.

THEORY

For a clear understanding of the design graphs, some theory beyond that presented in the first paper will be necessary, principally in the treatment of heat loss.

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† This paper is based on the M.Sc. thesis of W. C. Moffatt in the Department of Mechanical Engineering of Queen's University, Kingston, Ont., Canada.

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** Assistant Professor, Department of Mechanical Engineering, Queen's University. Presently Professor, Department of Engineering, Assumption University of Windsor, Essex College, Windsor, Ont., Canada. Associate Member of ASHRAE.

[1] Exponent numerals refer to References.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N.Y., June 1959.

The efficiency drop due to friction is given by $K\rho_1 V_1^2/2\Delta\rho g H$ where K may be evaluated as follows:

$$\text{for } N_{Re} > 3100, \quad K = f\left(\frac{L}{d}\right) + 52f^{1.60};$$

$$\text{for } N_{Re} < 3100, \quad K = f\left(\frac{L}{d}\right) + 25\left(\frac{H}{d}\right)^{0.6}$$

The expression for chimney efficiency[2] is

$$\frac{D}{\Delta\rho g H} = \frac{\Delta\rho_m}{\Delta\rho}\left(1 - \frac{K\rho_1 V_1^2}{2\Delta\rho g H}\right). \quad \ldots\ldots\ldots \quad (1)$$

This expression accounts for heat loss from the chimney if $\Delta\rho_m$ is the density difference $(\rho_o - \rho_m)$ where ρ_m is the density corresponding to a mean temperature T_m. This mean temperature is that temperature which if it existed throughout the chimney would result in the same draft condition as would occur if the gas entered at T_1 and left at T_2.

The case of the thin-walled metal chimney in which the wall has negligible resistance to heat flow will be considered first. For this case, the inside and outside surface films account for the only resistances to heat flow. The inside film coefficient may be evaluated[3] from the expression

$$\frac{h_i d}{k} = 0.023\left(\frac{dG}{\mu}\right)^{0.8}\left(\frac{C_p\mu}{k}\right)^{0.4}$$

Using air or flue gas within the limited temperature range under consideration, $C_p\mu/k = 0.74$ with little error. Therefore the expression reduces to

$$\frac{h_i d}{k} = 0.0204\left(\frac{dG}{\mu}\right)^{0.8}$$

or

$$h_i = 0.0204\left(\frac{k}{d}\right)\left(\frac{dG}{\mu}\right)^{0.8}$$

The conductivity for air is taken as

$$k = 0.0133 + 2.39 \times 10^{-5}\, T_g$$

where T_g is the flue gas temperature.

For the purposes of this analysis, the gas properties may be evaluated at the chimney inlet temperature with sufficient accuracy. The absolute viscosity was evaluated at 500 F and assumed constant at 0.0679 pounds per (foot)(hour).

The resulting equation for the resistance term would be

$$R_i = \frac{1}{h_i} = \frac{d^{1.8}}{(2.71 \times 10^{-4} + 4.88 \times 10^{-7} T_i)(18.8w)^{0.8}} \quad \ldots\ldots \quad (2)$$

The outside film coefficient for heated vertical flues[3] is given as

$$h_o = 0.29\left(\frac{T_s - T_o}{d}\right)^{0.25} \quad \ldots\ldots\ldots \quad (3)$$

The surface temperature T_s may be evaluated from

$$T_s = T_i - \frac{R_i}{R_i + R_o}(T_i - T_o)$$

NOMENCLATURE

Symbol	*Quantity*	*Dimensions*
A	= area	L^2
C_p	= specific heat at constant pressure	$H/M\theta$
D	= draft	M/LT^2
d	= diameter	L
od	= outside diameter of chimney	L
G	= mass flow rate per unit area	M/TL^2
g	= acceleration due to gravity	L/T^2
H	= height	L
h	= heat transfer coefficient	$H/TL^2\theta$
k	= thermal conductivity	$H/LT\theta$
L	= total length	L
R	= heat transfer resistance	$TL^2\theta/H$
$\rho, \Delta\rho$	= density, density difference	M/L^3
T	= temperature	θ
U	= overall heat transfer coefficient	$H/TL^2\theta$
V	= velocity	L/T
μ	= absolute viscosity	M/LT
w	= weight flow	M/T
X	= UL/C_p	
Y	= e^{UAL/wC_P}	

Dimensionless Groups

K = Euler's number = isothermal friction factor

f = Darcy's friction factor

N_{Re} = Reynold's number

Subscripts

$_i$ = inside

$_1$ = chimney inlet

$_2$ = chimney outlet

$_o$ = outside

$_s$ = surface

$_m$ = mean

Dimensions

The M L T θ H system of dimensions is used in this paper. The symbols and units are as follows:

M = mass in pounds.

L = length in feet.

T = time in seconds.

θ = temperature, Fahrenheit, degrees.

H = heat in Btu.

Combining these 2 equations the relationship may be stated

$$\frac{(R_i + R_o)}{R_o^5} = \frac{0.0071}{d}(T_i - T_o) \quad \ldots\ldots\ldots \quad (4)$$

The value of the mean chimney temperature T_m can be found[2] from

$$T_m = T_1 - 0.65\,(T_1 - T_2) \quad \ldots\ldots\ldots \quad (5)$$

T_2 must therefore be evaluated in order to use this expression.

$$dQ = wC_P\,dT_i = U\,\Delta T\,da$$

$$\int_{T_1}^{T_2} \frac{dT_i}{T_i - T_o} = \frac{Ua}{wC_P} = \frac{UAL}{wC_P}$$

Therefore,

$$\ln\left(\frac{T_1 - T_o}{T_2 - T_o}\right) = \frac{UAL}{wC_P}$$

$$\frac{T_1 - T_o}{T_2 - T_o} = e^{UAL/wC_P} \quad \ldots\ldots\ldots \quad (6)$$

The U value in this equation is for the chimney wall and in the special case of the aluminum wall

$$U = \frac{1}{R_i + R_o} \text{ where } R_i \text{ and } R_o$$

have been determined. The values of T_2 as calculated agreed with experimental values in all cases. The maximum difference noted was 8 percent with the majority being within 2 percent.

The mean temperature T_m can then be calculated from Equation 5.

For the case of a thick-walled, *e.g.* masonry, chimney the resistance to heat flow of the wall must be considered. In addition, when the wall thickness is of the same magnitude as the inside dimensions of the chimney, a shape factor is normally introduced and the normal conduction equation becomes

$$q = S\,k\,\Delta T$$

For a rectangular flue

$$S = \frac{P}{\Delta x} + 2.16 \text{ ft per ft of length}$$

and for a circular flue in a square chimney

$$S = 2 \times 10^{(1.14d/b)}$$

where

d = flue diam, and
b = length of one exterior side of chimney

It may be readily found then that

$$U_i = 1 \Big/ \left(R_i + \frac{A_i R_o}{A_o} + \frac{A_i}{Sk}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where

U_i = the overall heat transfer coefficient to be used with the inside surface area.

As was seen previously R_o could only be found after R_i was known. The inside resistance now used is

$$R_I = R_i + \frac{A_i}{Sk} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

This value for R_I is then used in the calculation of R_o.

Equation 6 is again used to determine T_2 with $U_i \times A_i$ replacing $U \times A$. Once having T_2 the value of T_m may be found as before. Then T_m is used to calculate the mean density ρ_m and from this $(\rho_o - \rho_m)$ is found.

This completes the theoretical treatment of chimney losses, and makes possible a relatively simple and reliable method of evaluating performance. In order to further simplify the design, graphical solutions of the foregoing equations have been developed.

GRAPHICAL SOLUTION OF CHIMNEY PERFORMANCE

In starting the design of a chimney, the following factors are normally known: flue gas temperature, T_1; chimney height, H; required draft, D_R; flue gas weight flow, w.

In addition to these there are other factors, the values of which must be assumed. For the development of the graphs it was assumed that the ambient air temperature T_o was 60 F, and the relative roughness of the flue was 0.001.

From the experimental results it was seen that the chimney efficiency reached a maximum at or near $(K\rho_1 V_1^2)/(2\Delta\rho gH) = 0.1$. For purposes of design it is desirable to choose conditions which will approximate this point of maximum efficiency. Fig. 1 is the solution of this expression and from it the chimney inside diameter may be found. Rectangular equivalents may be used in place of the round size as found. Fig. 1 was plotted assuming the flue gas temperature at the

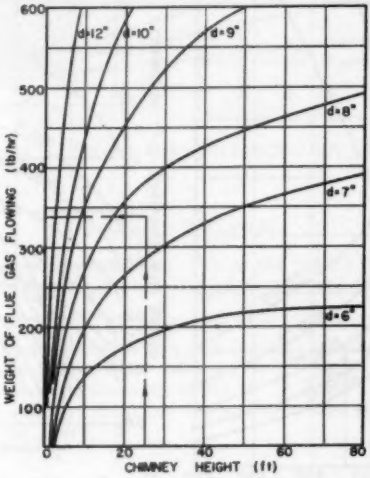

FIG. 1. . . .GRAPH FOR DETERMINATION OF OPTIMUM CHIMNEY SIZE

chimney inlet was between 400 and 600 F. However, this same graph may be used with temperatures between 300 and 700 F with acceptable accuracy.

Step 2 of the calculation is the determination of the actual draft which will be available in this size of chimney under the given conditions.

The actual chimney efficiency may be found from Equation 1. This equation however, reduces to

$$\frac{D}{\Delta\rho gH} = \frac{\Delta\rho_m}{\Delta\rho}(1 - 0.1)$$

or

$$D = 0.9\,\Delta\rho_m\, gH$$

$\Delta\rho_m$ is found by the method outlined in the theory.

Fig. 2 gives the solution of Equation 2 and yields a value for R_1, the inside film resistance to heat flow. For a chimney with a wall having an appreciable resistance

to heat flow, the total inside resistance term must be calculated from Equation 8. The thermal conductivity of the chimney material may be found from the normal sources of heat transfer data. Having found R_I, Fig. 3 may now be used to give the solution of Equation 4. This gives the value of R_o, the outside film resistance.

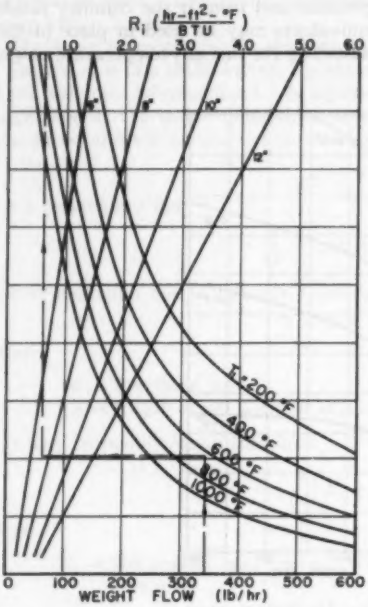

FIG. 2. . . .GRAPH FOR DETERMINATION OF INSIDE FILM RESISTANCE R_I

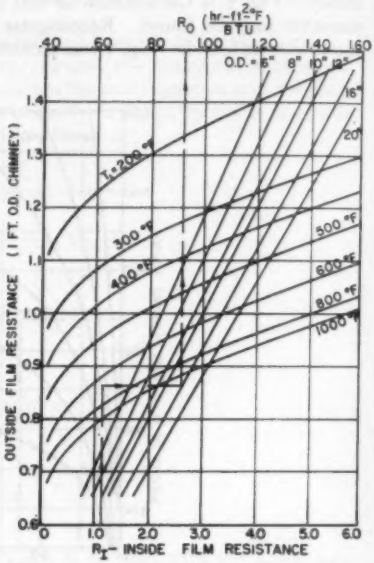

FIG. 3. . . .GRAPH FOR DETERMINATION OF OUTSIDE FILM RESISTANCE R_o

Using these values of R_I and R_o, U the overall heat transfer coefficient may be calculated from

$$U = 1 \Big/ \left(R_I + \frac{A_i R_o}{A_o} \right)$$

Using Figs. 4, 5 and 6 the values of X, Y and T_2 may be found and from Equation 5, T_m may now be calculated.

The actual draft for the given chimney is then determined from

$$D = 6.86\, H \left(\frac{1}{T_o} - \frac{1}{T_m} \right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

where the temperature T is in absolute degrees or $T = F + 460$, and draft, D, is in inches of water.

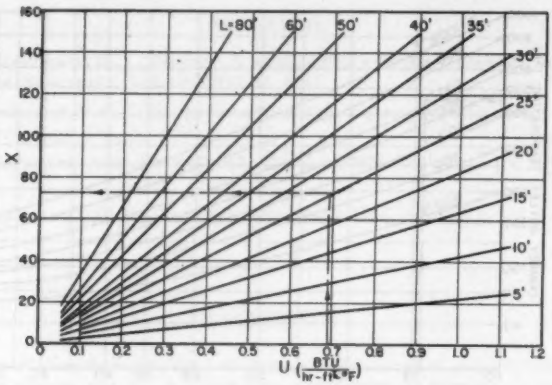

FIG. 4. . . .GRAPH FOR DETERMINATION OF X

SAMPLE SOLUTION

Example: Given fuel rate, 16 lb per hr of oil; CO_2 in flue gas, 10 percent; chimney height, 25 ft; chimney construction, common brick; flue gas temperature, 650 F.

Solution: Flue gas flow rate = 340 lb per hr from THE GUIDE[4]. From Fig. 1 the chimney diameter for maximum efficiency is seen to be between 7 and 8 in. The smaller diameter is chosen since it results in a lower efficiency drop than the selection of the larger size. This can be seen from performance curves.

Allowing for liner thickness, mortar and bricks, the exterior dimensions of the chimney will be 16 × 16-in.

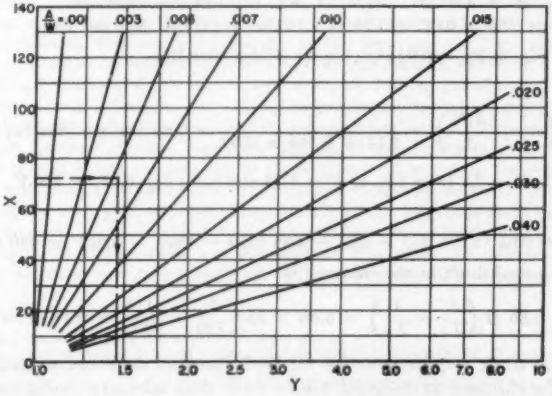

FIG. 5. . . .GRAPH FOR DETERMINATION OF Y

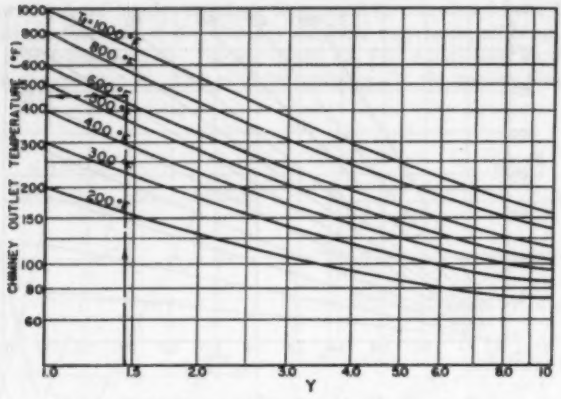

FIG. 6. . . .GRAPH FOR DETERMINATION OF EXIT TEMPERATURE T_2

To determine the chimney exit temperature, the following values will therefore apply:

Inside Surface Area per foot of length, $A_i = 1.83$ sq ft.

Outside Surface Area per foot of length, $A_o = 5.33$ sq ft.

Thermal Conductivity, $k = 0.48$ Btu per (hr) (ft) (F deg).

Shape Factor, $S = 2 \times 10^{(1.25 \times 7)/16} = 7.07$ ft.

$A_i/A_o = 0.344$

$A_i/Sk = 0.54$ (hr) (sq ft) (F deg) per Btu.

$A_i/w = 0.0054$ (sq ft) (hr) per lb.

From Fig. 2, $R_i = 0.60$ (hr) (sq ft) (F deg) per Btu and therefore

$R_I = R_i + A_i/Sk = 0.60 + 0.54 = 1.14$ (hr) (sq ft) (F deg) per Btu.

Then from Fig. 3, $R_o = 0.92$ (hr) (sq ft) (F deg) per Btu.

Therefore,

$$U = 1 \Big/ \left(R_I + \frac{A_i R_i}{A_o}\right) = \frac{1}{1.14 + 0.344 \times 0.92} = 0.69 \text{ Btu per (hr) (sq ft) (F deg).}$$

From Fig. 4, $X = 72$, from Fig. 5, $Y = 1.45$ and from Fig. 6, $T_2 = 450$ F.

Therefore,

$T_m = T_1 - 0.65\ (T_1 - T_2) = 650 - 0.65\ (650 - 450),\ = 520$ F $= 980$ Rankine

Hence the actual draft in the chimney is

$$D = 6.86\ H \left(\frac{1}{T_o} - \frac{1}{T_m}\right) = 6.86 \times 25 \left(\frac{1}{520} - \frac{1}{980}\right) = 0.15 \text{ in. of water.}$$

The normal draft requirement for oil-fired furnaces does not exceed 0.06 in. of water so the chimney as designed will be more than adequate under the given conditions.

REFERENCES

[1] W. G. Colborne and W. C. Moffatt: A fundamental analysis of chimney performance (ASHRAE TRANSACTIONS, Vol. 65, 1959, p. 417).

[2] W. G. Brown and W. G. Colborne: Fundamentals of chimney performance (*Canadian Journal of Technology*, Vol. 34, 1956, p. 354).

[3] W. H. McAdams: *Heat Transmission* (McGraw-Hill Book Co., Inc., New York, 1954).

[4] HEATING VENTILATING AIR CONDITIONING GUIDE 1958 (ASHAE).

DISCUSSION

W. G. BROWN*, Zurich, Switzerland (WRITTEN): This study of chimney performance by Professor Colborne and Mr. Moffatt helps considerably in showing up some of the weaknesses of old methods of chimney design which have been either too empirical or overly simplified. The paper itself is of particular interest to this discusser for two reasons:

First, the work represents, in essence, an extension of previous work by myself◘ in which much of the theory given in the paper was developed. This theory is also to be found elsewhere.† The auxiliary equipment described in the present paper is also that constructed by myself for the previous work.

Second, in the interval between the previous work and that of the authors another program of chimney testing‡ has been carried out at the Division of Building Research of the National Research Council of Canada which offers further information on the performance of masonry chimneys.

In particular, this new study has shown that the non-isothermal friction factor K does not depend only on the Reynolds number, but also strongly on the Grashof number, as in natural convection. This dependence on Grashof number was determined by comparison of K-values from isothermal model tests with the values of K determined with the masonry chimney under normal non-isothermal operation. In the masonry chimney the friction loss at low gas flow rates actually tended to become independent of the flow velocity. In addition, flow was turbulent no matter how low the Reynolds number. Only at high flow rates (corresponding to Reynolds numbers of 10,000 to 20,000) did the non-isothermal values of K approach those of the isothermal model tests. As a consequence of these tests it became clear that the previously reported† good agreement between the theoretical performance based on assumed isothermal friction and the performance of the masonry chimney tested by Achenbach and Cole was fortuitous. This means simply that it is not yet possible to properly extend test data for other sizes of chimneys as suggested previously and by Professor Colborne and Mr. Moffatt. In order to do this, the manner in which K depends on both the Reynolds and Grashof numbers must be known. Only for high flow rates can the design method suggested previously† be used. In the range of low to moderate flow rates the chimney efficiency varies sharply with the flow rate from a low value of perhaps 20 to 50 percent at near

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* Institut für kalorische Apparate und Kältetechnik der E.T.H.

◘ Brown, W. G.: An analysis of the performance of some model chimneys (M.Sc. Dissertation, Queen's University, Kingston, Ontario, 1955).

† Brown, W. G. and Colborne, W. G.: Fundamentals of chimney performance (*Canadian Journal Technology*, Vol. 34, 1956, p. 354).

‡ Brown, W. G. and Wachmann, C.: Draft performance of chimneys. (Submitted for publication).

zero flow to a peak efficiency of 70 to 90 percent. Hence errors in the value of K can cause large errors in estimating the efficiency for other chimney sizes. Since the flow rates corresponding to maximum efficiency are of greatest interest in chimney design, care must be exercised in interpreting this range.

Did the authors make any attempt to instrument their chimneys for determination or estimation of the actual friction losses during non-isothermal flow? If not, perhaps it would be permissible to suggest that some work of this kind be carried out before an attempt is made to offer a design method based on the use of isothermal friction factors. The model chimneys should be well suited for an investigation of this type to determine the general dependence of K on the Reynolds and Grashof numbers together.

A. G. WILSON, Ottawa, Ont., Canada: It is pleasing to see this further development of methods for predicting chimney performance. There is just one comment. In the experimental work and analysis steady state operation of the heating appliance was assumed. In any domestic heating system cycling operation of the unit will occur. A rigorous method of design of chimneys for domestic applications must ultimately take into account the variations in draft during non-steady operation of the appliance.

D. W. LOCKLIN, Columbus, Ohio: There are several points which stem from an interest in the application of this type of data to residential installations. Mr. Wilson has already mentioned the transient effects.

Attention should also be called to the fact that the critical period in a residential unit is usually during the initial start-up from a dead-cold condition of no draft. Repeated cycling at short and reasonably uniform intervals, as may be provided by many modern control systems in severe weather, perhaps approaches a steady-state condition of heat transfer in the chimney but the critical condition remains the infrequent cold start.

Another point worthy of mention is the variable effect of the barometric draft control on oil-fired equipment and the draft hood on gas-fired equipment. The mass flow rate up the chimney is difficult to predict because of the dilution effect which reduces stack temperature and also alters the draft.

In an oil-fired installation with a barometric draft control, there is the variable starting behavior which alters the venting system in addition to the normal starting transient condition. Some such controls are adjusted to be partly open in this zero draft condition and others are adjusted to be closed, but provide some clearance for leakage; in either case, the control blade usually will open as draft is established to bleed air from the furnace room into the chimney.

Another variable has to do with the effect of wind, not only on the chimney draft by aerodynamic action at the chimney top but on the heat transfer through the chimney wall. This affects the outside film coefficient for heat transfer on an exterior chimney face differently from an inside face.

The authors have done an excellent job in analyzing the effect of many variables in the already difficult steady state condition for chimney performance. The additional points just raised result in further complications to the analysis. However, it is hoped that the authors will extend their investigation to provide some quantitative information concerning these factors, and possibly to compare their analysis with field performance under these conditions.

AUTHORS' CLOSURE (Mr. Colborne): The authors thank Mr. Brown for his comments, which appear to be excellent and should prove very useful for anyone continuing the research in this field.

The equipment which Mr. Brown mentioned as being built by himself was rebuilt to a large extent. Certain features in the original equipment led to inaccuracies and difficult conditions under which to make readings. A new fan, a new entrance section to the air

measuring station, and a new air measuring station were installed. The outlet from the furnace to the chimney was completely redesigned in order to cut down on the stratification which originally occurred. This new outlet section took air from the furnace in a horizontal plane and, therefore, assured reasonably uniform temperatures. Turning vanes were designed to channel this constant temperature air evenly across the entrance section to the chimney.

The K-value which Mr. Brown mentioned is one of the subjects about which the authors are not satisfied. It was assumed that K was dependent on the friction factor. The basis of this assumption may be shown by taking the case of an ordinary elbow. The K for the ordinary elbow is a function of Reynold's number, the roughness ratio e/D, and radius ratio R/D.

In a straight pipe the K is a function of Reynold's number, e/D, and L/D. In attempting to give one K for an entire chimney it was necessary to combine the elbow at the chimney entrance with the straight pipe of the chimney stack. It was therefore assumed that K would be a function of Reynold's number, e/D, and L/D, eliminating the term R/D since in all cases it would be a sharp edged tee or elbow. Friction factor, f, on the other hand is a function of Reynold's number and e/D. Therefore, it was assumed that the K would be a function of friction factor, f, and L/D. This analysis was all based on the isothermal condition.

Inclusion of the Grashof number into this analysis is certainly very interesting and would require considerable additional study. The measure of friction loss in a chimney for the non-isothermal condition would in itself present many problems. It is hoped that someone can continue this work and do the required studies to determine the effect of Grashof number.

Mr. Brown made the point that the chimney efficiency curve took a sudden drop in the high efficiency region. The operating point which was selected for the chimney design work was 0.1 on the X-axis. Referring to any of the chimney performance curves shown in the paper it can be seen that any sudden drop in efficiency occurs well to the left of 0.1. It is the feeling of the authors, therefore, that unless the value of K is altered rather drastically it will have little effect on the chimney efficiency.

Mr. Wilson is correct in assuming that only steady state conditions have been considered. In the case of a domestic heating system cycling operation would occur. It is the intention, if the authors are able to continue this work, to investigate the problem of cycling operation. Where the chimney inlet temperature is varying with time then the mass of the chimney itself would have to be considered. The temperatures, both of the gases and of the chimney itself, would vary with time. The temperature variations that would occur would depend on the heating capacity and the insulating value of the material used in the chimney.

The authors can do nothing but agree with everything Mr. Locklin said. There are many problems still to be solved before a complete understanding of chimney performance will be possible. The cold start mentioned is very likely the most critical condition. Also, as mentioned, there are uncontrollable weather conditions. Where the cycle has long *on-periods* and short *off-periods* the chimney would probably reach what could be considered a steady state operation. On the other hand, long off-periods would turn the problem into a transient state situation. The authors have done nothing about the problem of cold starts. The model aluminum chimney would not likely be suitable for a study involving varying temperatures. This situation would involve the mass of the material making up the chimney. In the case of an outside chimney, the outside film coefficient would be influenced by wind. It is possible that the variations would be small enough to neglect, on an overall analysis.

Mr. Locklin also mentioned the effect of barometric draft control. The barometric damper will allow basement air or ambient air into the smoke pipe. The barometric damper in all cases cuts down the draft to a value suitable for the appliances being served.

This assumes that the draft produced by the chimney is greater than required by the appliance. For this reason the authors' design method works to the optimum point or the point of highest efficiency. The resulting chimney will produce, therefore, a higher draft than is normally required. The use of some draft reducing mechanism such as a barometric damper has been assumed, but the original calculations will be upset since outside air is admitted to the smoke pipe. However, the draft available at the heating appliances will be as required.

1677

No. 1677

INVESTIGATION AND CONTROL OF REFRIGERATOR NOISE

By E. A. Baillif* and J. P. Laughlin**, Evansville, Ind.

MODERN HOUSEHOLD APPLIANCES probably are receiving more attention now from the standpoint of noise than ever before. Their increasing numbers contribute materially to this. They are becoming more and more complex, with more and more moving parts, and inherently more efficient in many ways. Also, unless checked, the trend is toward more noise. It is a never ending battle to keep this noise within acceptable limits and of all appliances, the battle rages more furiously over the refrigerator than any other appliance, probably because its main job is to be cold, something which has no noise-producing connotations to anybody. Also it does not require an operator once it is plugged in, so it is expected to do its job unnoticed. Due to a moderately successful waging of this never ending battle, most of them do their job unobtrusively. This paper describes the processes involved in the pursuit of the solution to the many, many problems associated with refrigerator noise control.

Tools for Testing

The first essential, even if the *instruments* are ears, is a sound test area. This, for some evaluations, might be a quiet corner of the laboratory with makeshift walls or heavy curtains to form a "listening booth." But this becomes quite inadequate the moment one attempts to examine noise on other than an aural basis. For investigation of noise contributions in most frequency bands, and for establishing a history of reproducible data, a specially isolated room is a basic requirement. This room may be either anechoic (free field) or reverberant (diffuse), but one which has some of the characteristics of both is probably the most common.

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This compromise room is the so-called "padded cell" and is anechoic at notably high frequencies and essentially reverberant at low frequencies.

In almost all cases where the laboratory is in an industrial environment, a double wall, double door room is necessary. The inner room is suspended on either springs or rubber to isolate all frequencies above about 4 cps. This is a fairly practical limit and if a room is designed to isolate much below this, it becomes *wobbly* and uncomfortable in which to work.

The shape of the inner room is a factor, due to standing waves, particularly in the case of the *padded cell* type structure. Generally speaking, a rectangular

FIG. 1. . . .ENGINEERING SOUND ROOM

room is used and to provide a good standing wave ratio, i.e., the most diffuse field, its dimensions should be in a ratio of 3-4-5.[1] Other room shapes can be developed which give optimum diffuse field, but this leads to expensive construction.

In the facilities here described there are at present 3 sound test rooms available for product engineering test of refrigerators or their components, all of the semi-reverberant type. One of these is suspended on springs but does not have an enclosing wall around it. It is unsatisfactory in the high laboratory ambient noise environment because of wall panel noise impingement and serves basically as a *listening room*. It is interesting to note that with this room's extremely flexible mounting (1.6 cps) it is quite an uncomfortable place to work, producing something akin to mild seasickness.

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[1] According to Dr. H. F. Olson, RCA Laboratories.

A double wall room consisting of one padded cell and one reverberant test cell is under construction presently. One part of this, the padded cell, is complete and in operation and is shown in Fig. 1. A schematic of the complete arrangement is shown in Fig. 2. It can be seen that the instrumentation is outside the suspended test chamber, a good arrangement, because otherwise the instrument noise may interfere with the test in progress. However, this has the disadvantage of removing the operator from the *feel* of the object on test.

Special attention must be paid to the openings in the rooms. Each wall should have its own door and both of these doors must have an effective door gasket seal. All other openings such as those for power, lighting, and instrument cables, should be well caulked. Any narrow aperture will create an acoustic leak particularly at high frequencies.

Temperature control of the test cell and instrument areas is desirable both from the standpoint of operator comfort and for minimizing the temperature

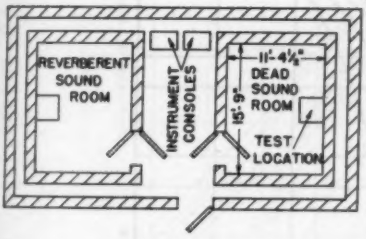

FIG. 2. . . .SCHEMATIC DIAGRAM OF SOUND ROOMS

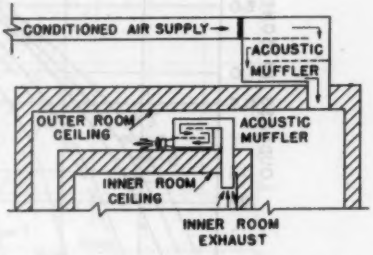

FIG. 3. . . .SCHEMATIC DIAGRAM OF AIR-CONDITIONING SYSTEM IN SOUND ROOMS

effect on the instruments and the operating noise of the test object. The least expensive method is to provide a direct expansion, fan cooled refrigerated coil in the test room with an external refrigeration system, which is then pumped down and the fan turned off for complete absence of noise during test. This type system functions during non-test periods with the room doors open to provide ventilation. Fig. 3 shows the arrangement for air conditioning the room in Figs. 1, 2, and 16 which utilizes an air-conditioned outer room with duct treatment to minimize noise carry through, and with a baffled low velocity conditioned air system continually changing the air of the test cell.

In this test room the practice has been to endeavor to make the refrigerator perform noise-wise in a manner similar to that encountered in a home. To this end, a solid floor is usually used under the refrigerator on test. Generally, one wall at least is also hard surfaced to give the effect of the reflective plaster or tile kitchen wall, while the remainder of the interior surface of the test cell is treated with acoustic material to provide an environment with controlled absorption.

The science of sound is not a simple subject and its study must involve not only physical factors, but psychological factors as well. This is because the human ear is one of nature's more complex structures, and not only functions as an extremely sensitive microphone, but also as a selective analyzer with the ability to extract

pitch, loudness and *quality* of noise in the presence of interfering background noise.

The task then of finding an instrument to make physical measurements of noise in some manner so that some of the human reactions to it can be classified, would

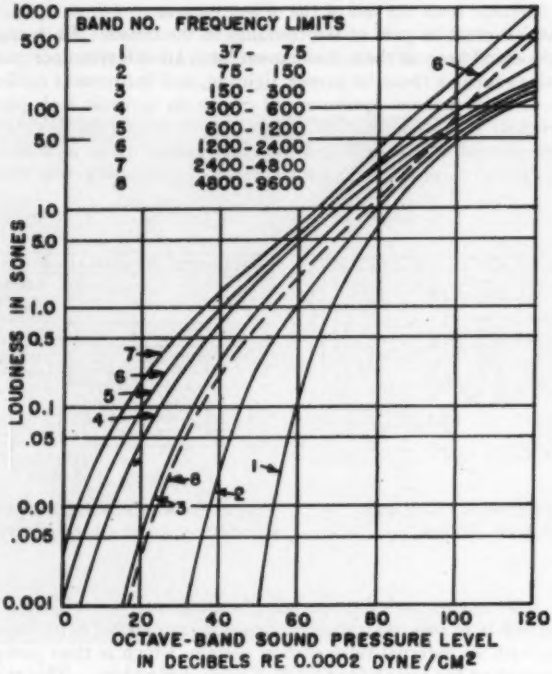

FIG. 4A. . . .CHART FOR CONVERTING FROM OCTAVE BAND LEVELS IN DB TO LOUDNESS IN SONES (AFTER BERANEK AND PETERSON)

seem a formidable one. Besides this, the factors mentioned are interdependent, and also are affected by the experience of the listener.

There are basically 2 objective properties of noise which are measurable. These are sound-pressure level and frequency, and relate subjectively to loudness and pitch. Therefore, to attempt to instrumentate and achieve the measurable objective properties of noise there must be provided a transducer to pick up air-borne or structure borne vibrations since noise has its source in vibration, that is, a microphone or vibration pickup; an amplifier to transform the output of the transducer into a useful signal level; a means of controlling the signal both as to intensity and choice of frequency response, namely, an attenuator and weighting

networks; a means to segregate out certain specific frequency ranges for detailed analysis; and a means of indicating or recording the output signal.

It has not been too long since the measurement of sound pressures was a complex task involving the use of classical physics techniques. The first commercial measuring instruments were complex and bulky. The progress made in recent years in the field of noise measuring apparatus has been phenomenal. These

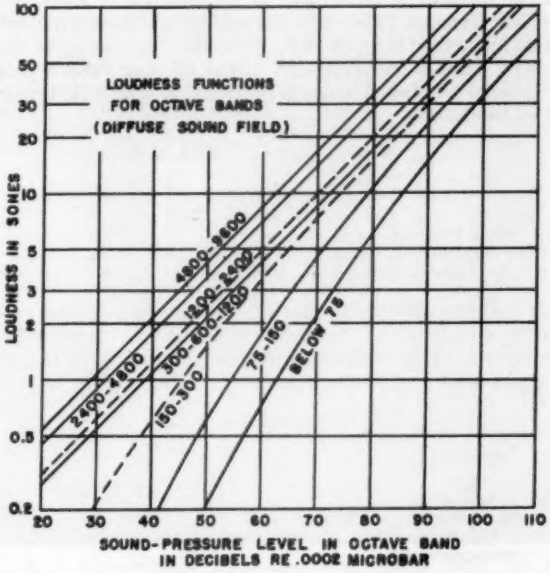

FIG. 4B. . . .CHART FOR CONVERTING FROM OCTAVE BAND LEVELS IN DB TO LOUDNESS IN SONES (AFTER STEVENS)

modern instruments are described in many places in the literature and will not be discussed further here.

INTERPRETATION OF RESULTS

Almost universally, when *noise measurement* is mentioned, the immediate thought is *decibels*. Actually, the use of a single value as a rating for the loudness of a noise has a serious drawback, namely that decibel readings have little relation to the *loudness* of a noise as *heard* by the ear. This is because the ear's response is not flat with respect to frequency, or in other words it tends to evaluate different frequencies differently. Although this is partly taken into account by weighting networks on the sound-level meters, it is not nearly accurate enough to make the resultant db readings represent a rank order of loudness.

Much more satisfactory is a rating system based on loudness in units of sones which does not pre-suppose similar frequency distribution. To obtain the sone value for a sound, it is first necessary to feed the flat network signal from a sound-level meter into, for example, an octave-band analyzer, which gives a db reading for each of a series of frequency bands. From a loudness *vs* octave band sound-level chart, can be obtained the sone value for each frequency band, and by then applying a simple formula, the overall loudness in sones for the particular noise in question can be obtained readily.

There are two methods of performing the computation of sones, each based on its own chart and formula. The one which has been in use for a number of years and is still a useful method is that which utilizes the chart in Fig. 4*a* which is based on work done by Beranek and Peterson[2]. After the sone value is determined for each band the total number of sones is obtained by simply adding together the loudness for all bands, or:

$$N_t = N_1 + N_2 + \cdots + N_8 = \Sigma_N$$

where

N_t = overall loudness
N_1, N_2 = loudness in each band
Σ_N = sum of all loudness for all bands

The other method has been rapidly gaining favor in the literature and in practice and is from the work of Dr. S. S. Stevens of the Psycho-Acoustic Laboratory at Harvard University. This consists of using the chart shown in Fig. 4*b*[3], and than applying the formula:

$$N_t = N_m + F(\Sigma_N - N_m)$$

where

N_t, Σ_N are as before
N_m = the loudness of the loudest band
F = a factor depending on measurement conditions (0.3 for octave band measurements).

It can be seen that it is possible by either method to have a sound with a lower decibel reading than another, be the louder of the two. This is quite often true when one sound has strong high frequency components as compared to another which has strong low frequency components. The one with the predominant high frequency will generally be the louder even though the db ratings may be equal or even reversed.

METHOD OF TESTING COMPRESSORS

Since the modern refrigerant compressor is basically a high speed electro-mechanical device, noise reduction measures would be expected to and do require the use of many techniques involving vibration isolation, pulsation mufflers, flexible connecting tubing, and component balancing. However, due to the

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[2] Willem Brand and J. T. Broch, *Bruel and Kjaer Technical Review* (Brush Electronics Co.) No. 4, 1955, Fig. 4.
[3] S. S. Stevens, *Noise Control*, September, 1957, pp. 3, 11.

thermodynamics involving relatively large pressure and temperature variations, oil refrigerant mixtures, and the miniaturization of components, there are other special techniques demanded over and above those encountered in noise studies involving a less complex mechanism.

For example, where the noise reduction is concerned with the metallic parts of the mechanism, one must consider surface finish, clearances, and assembly relationships. Although these are held to extreme tolerances and materials must always

FIG. 5. . . .COMPRESSOR ARRANGED FOR NOISE TEST ON A SPRING ISOLATOR

meet carefully established specifications, they can still be a factor in the resultant noise when altering the design to accommodate a different unit or to effect cost reduction. Admittedly, the noise here is controllable by emphasis on quality of parts and assembly procedures, once the design is established, but the procedures involved in determining the economical limits on quality, as it affects noise, call for special investigative processes.

As another example, consider the influence of the refrigerant flow and its pulsations. Here the pulsating flow is affected by the resistance to flow in the tubing and by the conditions at both ends.

One method which is reliable but costly is the exhaustive method of testing each variation in design in steps to establish a trend or a breaking point. A method leading to the same result but with less time and effort is preferable. This is possible when one departs from the basic sound-level meter and endeavors to analyze the frequency components of the compressor noise. To accomplish this,

one can make use of auxiliary test apparatus including frequency analyzers, pressure transducers, magnetic excitation of components and possibly most importantly, the science of mechanical vibrations itself.

Of special interest, from a design standpoint, are resonant frequencies of machine components. These can sometimes be calculated but because of the complexity of the shape of the component and the manner in which it is attached to the system this usually is difficult if not impossible. However, on a sample part, the resonant frequencies can be determined by either striking it or driving it electromagnetically

FIG. 6. . . .QUALITY CONTROL NOISE TEST CELL FOR COMPRESSORS

while comparing the emitted tones with a variable audio frequency sound source. Resonant components are bothersome because they have families of resonant frequencies, namely, multiple harmonics, any one or several of which may be excited into forced vibration either by repetitive impacts or oscillatory exciting forces near resonance.

With all the defects that are possible, which can also influence performance, it is easy to see why high standards are set on the production of parts for compressors. Long experience and constant surveillance by quality control allow but few compressors to reach final compressor assembly with adverse noise, and it is at this point that a noise check is made. This is the so-called *free space test* wherein a compressor is run under normal operating conditions within a noise test cell. The *free space* noise test takes two forms as shown in Fig. 5 and Fig. 6. Fig. 5 shows the compressor arranged for noise test on a spring isolator in one of the product engineering sound rooms. The engineering test makes possible the analysis of new compressor designs and also the spot checking of production compressor noise. Fig. 6 shows the quality control noise test for compressors

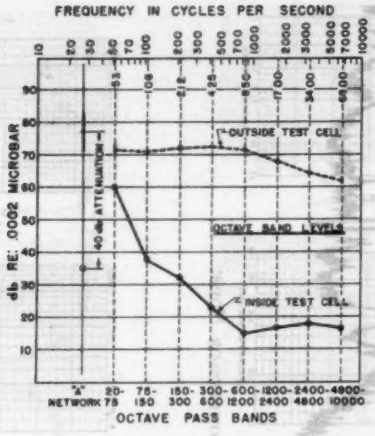

FIG. 7. . . .ATTENUATION CHART FOR QUALITY CONTROL NOISE TEST CELL

and is used as a gauge of production noise. The test cell shown in Fig. 6 was designed to produce a usable ambient sound level in a high industrial ambient noise level. Fig. 7 shows the results of this attenuation of ambient noise which makes possible the study of the significant compressor noise frequencies. Compressors whose noise levels exceed a standard are examined and the deviations

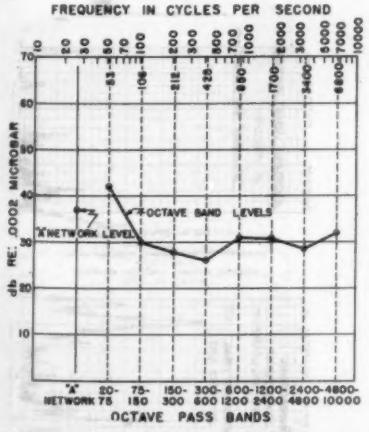

FIG. 8. . . .COMPRESSOR NOISE EXPRESSED IN DB

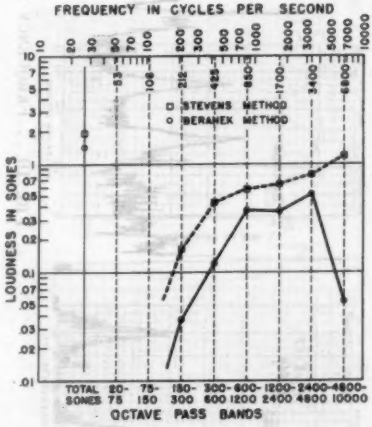

FIG. 9. . . .COMPRESSOR NOISE EXPRESSED IN SONES

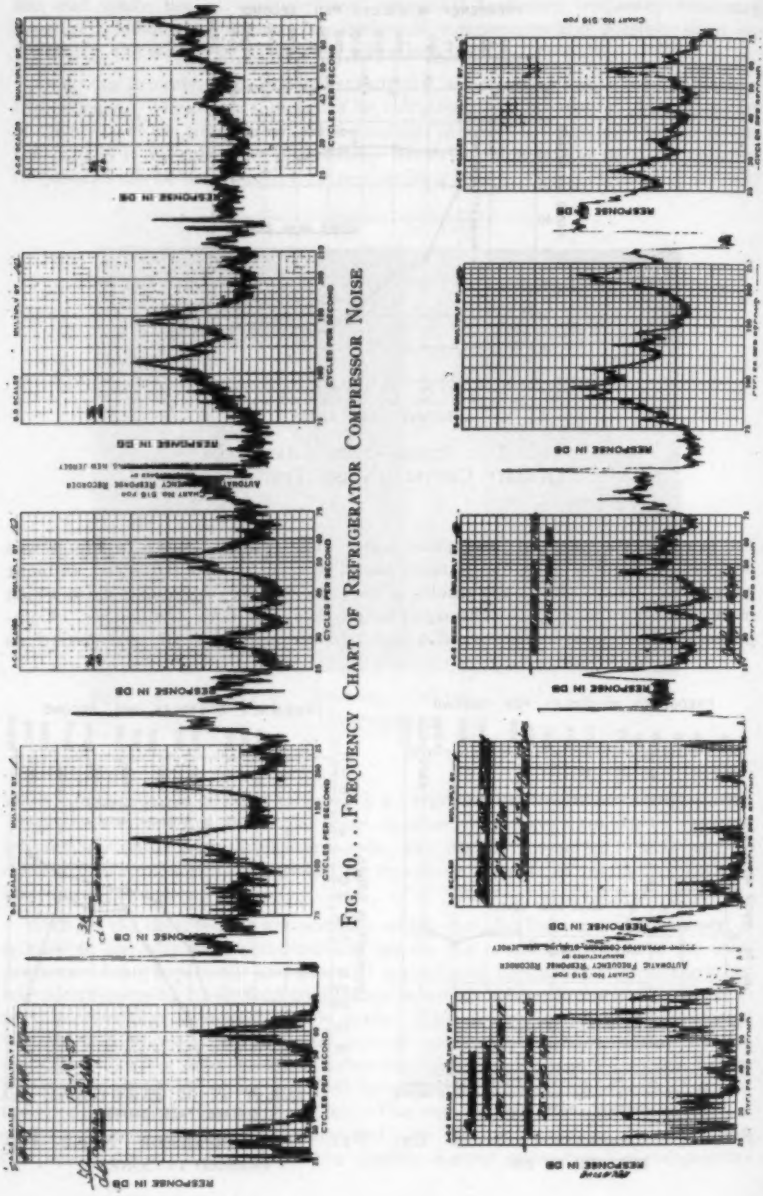

FIG. 10. . . .FREQUENCY CHART OF REFRIGERATOR COMPRESSOR NOISE

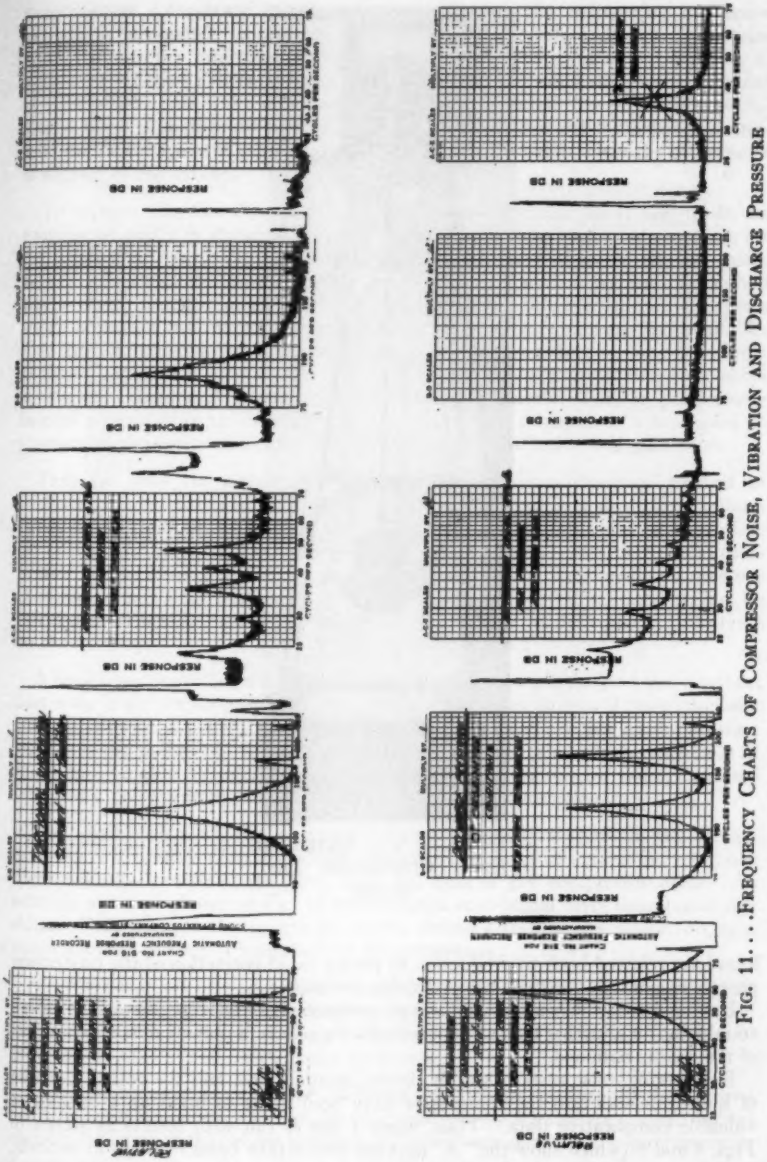

FIG. 11. . . .FREQUENCY CHARTS OF COMPRESSOR NOISE, VIBRATION AND DISCHARGE PRESSURE

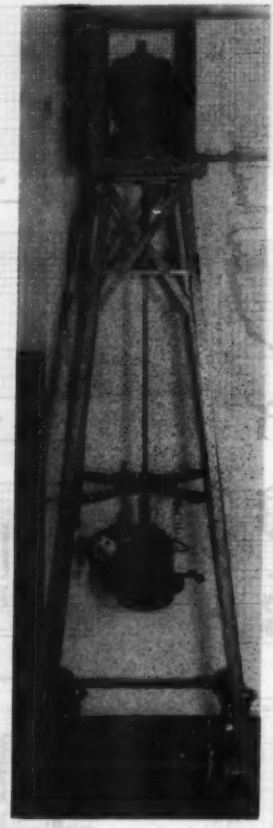

FIG. 12. . . .EXTERNAL DRIVE COMPRESSOR TEST STAND

found are referred back to production to permit rapid correction of the production process responsible for the noise producing deviations.

Having traced the compressor through production, let us retrogress to the design standpoint for a further look at the methods of analysis of noise, and some examples of the results obtained.

Engineering noise tests based on theory, assumptions, and the actual insertion of known defects into the compressor have been made to build up a backlog of valuable comparative data. Thus, when a test is run with results as shown in Figs. 8 and 9 (which show the "A" network and octave band readings in decibels, and the computed sone rating, by both the Beranek and Stevens methods, for an

experimental compressor tested under operation on the engineering free space set-up), it is possible to compare these data with the reference data from prior tests. By proper interpretations of these data it is to some extent possible to determine whether this compressor will perform satisfactorily from the standpoint of noise when applied to a refrigerator. It should be noted that proper interpretation is important since differences in the test setup and sound reduction measures used on the refrigerator cabinet alter certain frequency bands when the compressor is applied to the cabinet.

To further pinpoint the noise spectrum a frequency analysis is also made, an example of which is shown in Fig. 10. This is a discrete frequency spectrum for the same compressor shown on an octave band basis in Fig. 8. It can easily be seen how the discrete or individual frequencies can be identified by this method and a more detailed analysis thus made possible. Other frequency analyzers which present a continuous portrayal of amplitude *vs* frequency are also available.

A still further step can be made in the analysis by recording on tape the effects of either component or operating condition changes and then playing back short connected excerpts of these tapes for aural judgments. This is particularly helpful when delays are encountered in experimental changes, or it is desired to compare experimental designs *vs* production on an aural *before and after* basis.

Thus far only the airborne vibration transducer or microphone method of analysis has been discussed. There are other transducers which also will contribute valuable information on noise analysis, for example, the vibration pickup. Use of these in the torsional, vertical and horizontal aspects of vibration of the compressor can be helpful in the determination of component unbalance and pressure unbalances in the compressor. Compressor housing vibration patterns can be traced out with a low mass pick-up placed successively at multiple grid positions on the housing.

A device particularly valuable in diagnosing compressor noise and the resultant unit noise is the pressure transducer. Applied to the gas stream of the compressor this transducer can show the effects of internal component changes on the pressure wave and the before and after effects of mufflers. Fig. 11 shows a comparison of the data available by means of the mentioned test methods for noise, torsional vibrations, and pressure pulsation on an experimental compressor.

The possible tests have by no means been exhausted at this point as two further examples will show. One interesting method of analysis is that of isolating one disturbance by removing its effect from the field of the compressor noise. An example of this is shown in Fig. 12 which shows an external drive compressor test stand. This permits the compressor to be driven either by its motor, or by a motor coupled through a long shaft to the compressor.

A further example of a special test method is found in the use of magnetic pick-ups on a rotating part of the compressor in combination with a capacitive pickup formed from the discharge valve of the compressor, to investigate the effect of compressor valve action on the noise spectrum.

It should be apparent that the basic instruments are really only the starting point in analyzing noise and can be supplemented by special apparatus, limited only by the imagination of the engineer, to attack any phase of the compressor as a noise generator.

METHOD OF TESTING UNITS

The type of instrumentation chosen for a specific noise evaluation depends ultimately upon the purpose for which the information is to be obtained. The human ear still has its place in the evaluation of noise if the observer is trained for this purpose. A working example of this type of evaluation is shown in Fig. 13

FIG. 13. . . .QUALITY CONTROL ACOUSTICAL LABYRINTH FOR UNIT NOISE TEST

which is an acoustical labyrinth used on the production line to 100 percent check unit noise.

It is not feasible economically nor necessary to 100 percent noise test refrigerators at production rates common in refrigerator manufacture. However, potentially noisy units should be culled from the unit assembly line to avoid the cost of possible later rejection as a finished refrigerator, or a possible field service call.

The labyrinth serves this purpose by providing a low ambient noise level to enable a trained operator to aurally evaluate the merit of each compressor as it passes him on the moving conveyor chain.

One can appreciate well that in any system of this sort there will be some conveyor noise, rattles of evaporator tubing and similar noises, but the trained ear can easily discriminate against this extraneous noise as long as the ambient noise is held to a low level.

Note that, in the accompanying illustration, the operator can hear an abnormal noise as the unit rounds the bend coming toward him and is thus alerted to carefully scrutinize this unit for compressor noise.

It should also be mentioned that the ear is not infallible due to the human element, and in the case of a high percentage of rejected units they are rerun and

FIG. 14. . . .REFRIGERATOR IN ENGINEERING SOUND ROOM

listened to by a committee to determine whether the compressor assembly has a potential problem needing immediate attention, or whether the operator just had an off day.

The actual percentage of rejects due to this test is quite low by reason of the high standards set on compressor assembly and the aforementioned compressor noise test, but it serves as a double check on unit noise and promotes a high quality standard.

TESTING COMPLETE REFRIGERATOR

This is probably the most important test since now for the first time all the potential noise producing components are in place.

The first consideration of the overall refrigerator noise occurs in the design stage, then the engineering sound laboratory proves the design and it is placed in production. Here again the overall noise must be evaluated to insure that appurte-

nances, materials, assembly procedures, and like factors, are not contributing to offset the careful noise screening already accorded the compressor and unit.

These methods used to accomplish this in the industry are varied; ranging from visual observation of the components to 100 percent testing of all refrigerators by a method similar to that described previously for unit testing.

One method which has met the criteria of high quality at reasonable expense is to 100 percent visually check the refrigerator for assembly defects, test a quality control sample lot before crating, and a lesser number of samples from the warehouse. The latter test is sometimes conducted by the engineering sound laboratory

FIG. 15. . . INSTRUMENT CONSOLE IN QUALITY CONTROL SOUND ROOM

and permits including the processes of crating, transport, and uncrating, as would be the case in the store or customer's house.

Again, as previously mentioned, much can be said as to how to noise test the refrigerator; one method which has been time proven is hereinafter described. The actual testing arrangement is shown in Fig. 14, which is a view of a refrigerator situated in place in the engineering test room. It should be mentioned at this point that the quality control sound test room and the test method used are identical to that used by engineering except for the elaborateness of the instrumentation. The quality control sound room is shown in Figs. 15 and 16.

The refrigerator is run, usually outside the test rooms, for an interval to reach equilibrium conditions, before any sound readings are taken. This is done to achieve a more typical operation than would occur during a pulldown. Actually, however, a fairly good idea of the noise level can be had during a pulldown if a quick check is necessary.

The microphone and the measuring instruments comprise the balance of the test setup. The microphone is placed at a height of 48 in. from the floor, which is the approximate ear height of a seated person and at the 3 positions around the refrigerator, the 3 positions providing a means of arriving at a more average level.

Markings on the floor facilitate placement of the microphone. The instruments for reading and recording, referred to earlier, are located outside the sound test rooms as shown in Figs. 1 and 15.

Data from these tests are obtained from the various instruments as described here. The sound level meter provides "A," "B," and "C" network readings.

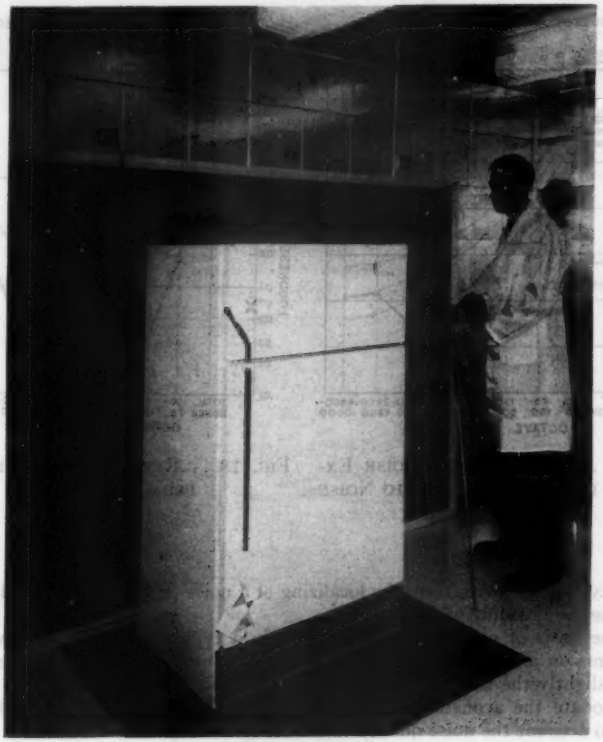

FIG. 16. . . .PHOTOGRAPH OF QUALITY CONTROL SOUND ROOM SHOWING REFRIGERATOR ON TEST

Background noise level corrections, if necessary, are made on the readings and the three "A" network readings arithmetically averaged and this figure recorded. For years this was the extent of a noise test in the industry and as mentioned before, as long as one refrigerator is compared to another both having similar frequency spectrums, this average decibel reading corresponds reasonably well to a loudness rating in sones. Sone readings, obtained as previously described, are now used, as

well as the noise criteria curves[4], but it has so far been impractical to completely eliminate the "A" network reading due to the long association with it by people who have only limited contact with the noise test process.

The noise criteria curves are rapidly gaining acceptance in the literature and have served as a basis for establishing a go-no-go criterion for the quality control sound room. Their particular advantage, from this standpoint, is the octave

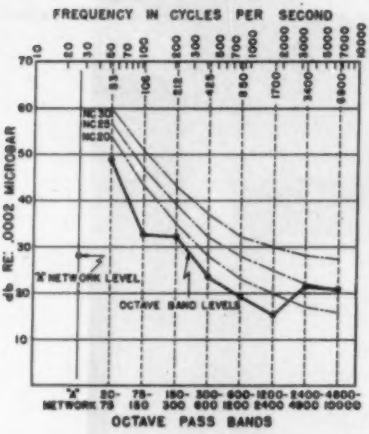

FIG. 17. . . .REFRIGERATOR NOISE EXPRESSED IN DB AS COMPARED TO NOISE CRITERIA CURVES

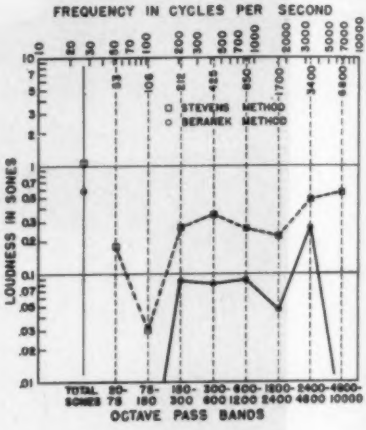

FIG. 18. . . .REFRIGERATOR NOISE EXPRESSED IN SONES

frequency display, permitting the localizing of a noise problem not possible by the one value sone reading.

On the basis of comparative tests made in a typical residence and two sound test rooms on a fairly large sampling of refrigerators, it was found necessary to modify slightly the noise criteria curves as published in the literature, in order to accommodate the acoustical environment of the sound test rooms, and also to take into account the noise quality of the refrigerators.

The effect of the latter can readily be understood if one considers the static type units in comparison to the forced draft models with their masking, although higher level fan-air noise.

To obtain the data necessary for the sone and noise criteria report, the output of the sound-level meter is fed into an octave-band or similar analyzer, for the 3 microphone positions; the decibel level in each band obtained is corrected for background noise, if necessary; and each octave band set of readings is arithmetically averaged. From this data the noise criteria plot can be made directly, and as

[4] L. L. Beranek, *Noise Control*, January, 1957, p. 19.

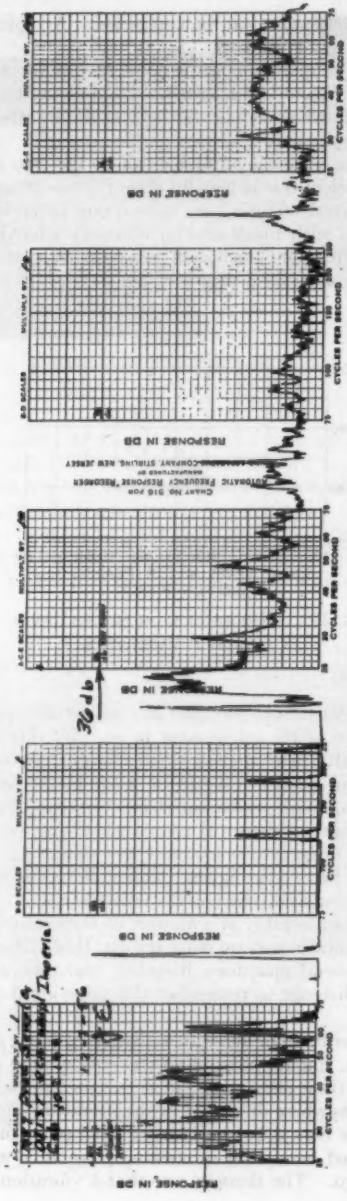

FIG. 19. . . .FREQUENCY CHART FOR REFRIGERATOR NOISE

previously described, a sone value can be computed. A typical set of data for a refrigerator is shown in Figs. 17 and 18.

This is the extent of the quality control noise test as far as obtaining and reporting of the individual datum is concerned. Engineering has, however, a somewhat broader function, and hence extends the test data gathering to include the use of some additional equipment.

Thus, the output of the sound level meter can be fed into a discrete-frequency sound analyzer and recorder; this is usually done for one or more positions of the microphone. Such a record enables one to examine individual frequencies and hence predict or theorize with much greater accuracy what is causing a specific noise. A typical chart from one such analyzer-recorder system is shown in Fig. 19. This is for the same noise as is shown in the form of octave bands in Fig. 17.

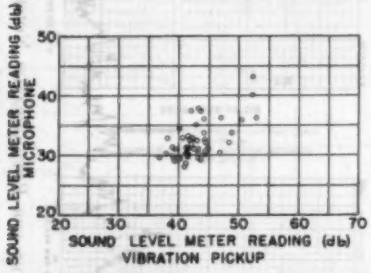

FIG. 20. . . .REFRIGERATOR SOUND LEVEL *vs* CABINET VIBRATION

Of a somewhat different nature but also of considerable value is the sound recording on magnetic tape of the refrigerator noise. By this method a design can be checked before and after a change is made; designs can be compared to competitive refrigerators; and a noise test can be made in the field for later analysis. The value of this test method, which supplements those described previously, is the obtaining of a permanent historical record which can be analyzed again at a future date should it be desired to re-evaluate any data. It should be noted that the tape system should be of the highest quality since the inaccuracy of the recorder, however small, is superimposed on the inaccuracies of the other instruments. A still further usage is the assembly of a number of tests recorded on tape to enable giving management an aural report on noise results; this without the usual problem of moving in, one or several machines, listening, and then after the normal disruptions expecting the listener to remember the prior aural effect while listening to other machines.

Again, there are several other devices which may have a special application such as the oscillograph, impact noise-analyzer, strobe lamp and vibration meter. Although use is made of these instruments in the engineering laboratory as occasions may dictate, they are elsewhere discussed in the literature and with one exception, to elaborate on their use here is beyond the intended scope of this paper.

The one case discussed here is an interesting possibility which was closely examined several years ago. The thought was that a vibration pickup, which could

FIG. 21. . . .SOLID (LEFT) *vs* RESILIENT FAN MOUNTING

be used on the assembly line might be applied at a predetermined spot on the compressor, unit, or cabinet, so located as to produce an indication which would correlate to the sound level of the machine. If this could be done an especially valuable means of controlling noise could be obtained. Several types of pickups were investigated and many different locations explored, but to little avail. There was some correlation in the most noisy machines which would have been singled out anyway, but the correlation did not exist to a useful degree for instance on a

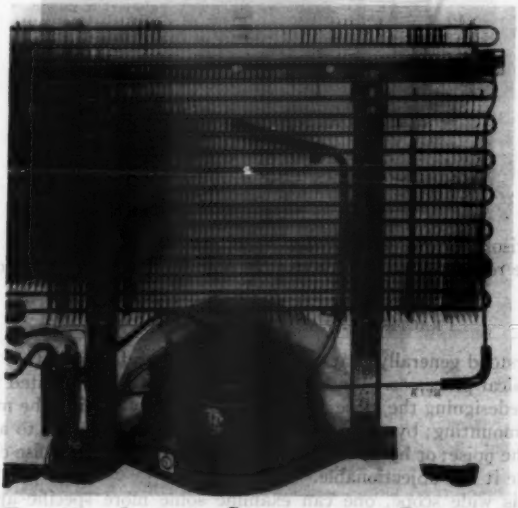

FIG. 22. . . .INTEGRAL CONDENSER AND PRECOOLER OF REFRIGERATOR

series of cabinet tests, as can be seen from Fig. 20. The failure of such a test method quite probably stems from the fact that the stresses and vibration patterns in the devices are not closely enough repeated from one to the next to provide a

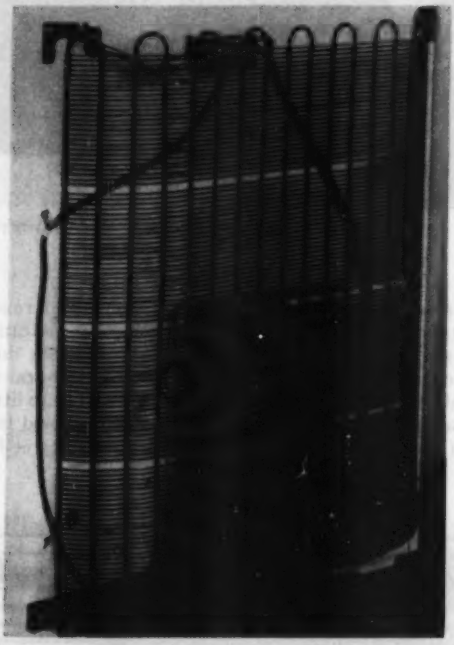

FIG. 23. . . .SEPARATION OF PRECOOLER AND CONDENSER IN AN OLD MODEL REFRIGERATOR

valid comparison. However, the fact remains that a very valuable noise control tool would be realized if some such device could be made to do the job.

PRACTICAL EXAMPLES OF NOISE CONTROL

It is understood generally that noise is generated and transmitted by a number of basic physical processes. And further, that it can be eliminated frequently or reduced by redesigning the noise producing devices; by placing the machine on an appropriate mounting; by adding mufflers, filters, or dampeners to absorb, block, or dampen the noise; or by modifying the device to change its noise characteristics so as to make it less objectionable.

Within this wide scope, one can examine some more specific applications of noise control as applied to refrigerators. The examples described here generally illustrate two avenues of approach: one where the same device or change can be

made to accomplish some additional desirable function, the other where a large significant noise reduction is possible. The guiding principle here is the problem of cost. The noise reduction which will not ultimately pay for itself in increased

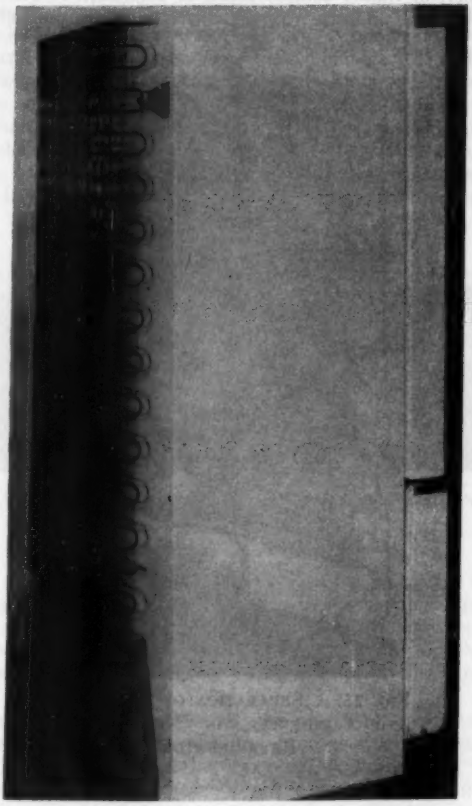

FIG. 24. . . .COVERED COMPRESSOR AND FLEXIBLE MOUNTED INTEGRAL PRECOOLER AND CONDENSER IN REFRIGERATOR

customer acceptance is not generally warranted. This creates a paradox of attempting to get something for nothing, since although the customer wants a quiet box he will generally not pay a premium cost to get it.

To achieve the desired result then, the engineer must be diligent in his interrogation of new designs, and modifications to accepted designs in order to maintain, or if possible improve the noise of the product.

With the advent of the modern automatic defrost, built-in refrigerator new problems of noise control arise as a result of the appurtenances necessary to achieve these desirable features.

Consider the interior fan which is variously located in the door, normal storage compartment liner, or freezer compartment, or combination thereof. The noise of this device should be unobtrusive, particularly with the unit off and the doors

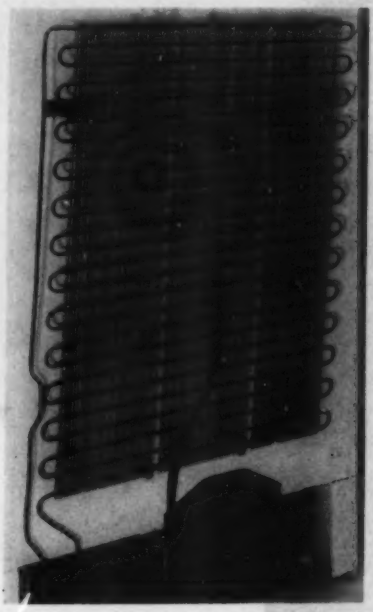

FIG. 25. . . .SEPARATION OF PRECOOLER AND CONDENSER IN NEW MODEL REFRIGERATOR

closed, otherwise the customer may erroneously believe the unit is running too much.

One noise control example on an interior fan resulted from being able to demonstrate the deleterious effects of the fan blade being too close to a louver system and evaporate plate edge. The resulting siren noise occurring at two distinct and pronounced frequencies was rectified by changing the relationships of the parts to alleviate the noise while at the same time producing no outward aesthetic change, or affecting the unit performance.

Another example of fan noise control concerns a highside system. Here a larger quantity of air is moved, necessitating a motor capable of transmitting objectionable vibrations to the cabinet panels, if not isolated carefully or dampened. Several

mounting methods were explored and a vibration isolation type which appeared satisfactory on a prototype basis was tried on a limited production basis. Variations in assembly both within the fan mounting and the cabinet structure soon showed that the mounting was not compatible to mass production without uneconomical secondary noise control measures. An alternate design which was based on a highly damped solid structure was substituted with the result of a much more consistent low-level noise contribution. The two types are shown in Fig. 21. This noise control example also typifies the advantage of quality control and engineering both working in a check and balance system to prove out a design.

Fig. 22 shows the unit highside construction of a model of several years ago, having a combination precooler and condenser mounted through rubber grommets to the back of the refrigerator. This model although having an acceptable noise level, had a poor noise quality, a tendency to *beat* two closely associated frequencies together. The frequencies in question were the gas pulsation frequency and a

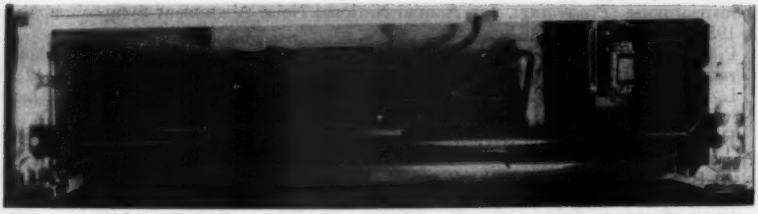

FIG. 26. . . .CONDENSER AIR BAFFLING USING ACOUSTICAL INSULATION

motor vibration frequency which differed by only a few cycles per second. Since the compressed gas was returned from the precooler into the compressor housing before being fed into the condenser, it appeared feasible to break the coupling of the two frequencies by mounting the precooler separately on the compressor. This was done as illustrated in Fig. 23. The success of this type of design is self-evident.

Another example on a static condensing system occurred more recently when it was desired to recombine the precooler and condenser. To maintain a suitable unitized condensing system, free of the beat noise, it was decided to provide for vibration isolation of the entire high side from the cabinet. To achieve additional noise reduction it was also decided to acoustically baffle the compressor noise. For this purpose a compressor cover was designed which very effectively reduced the noise level. The design is shown in Fig. 24. The flue effect of the cover resulted in two desirable side effects in that the compressor tended to run cooler, and water evaporation in the defrost drain pan increased as much as 40 percent.

For the final example of a static condensing system see Fig. 25 where the pendulum has swung back to the separate precooler and a solid mounted condenser. In this design additional noise reduction was achieved by breaking up the transmission path of the unit to the cabinet structure by multiple resilient isolation for low frequency noise, and acoustically treating the machine compartment for high frequency noise.

One final example typifies the *two for one* noise control measure where it was possible to achieve considerable high frequency noise attenuation by using acoustical sound insulation for the condenser air baffling as is shown in Fig. 26. It is of interest to note here that the residual noise of this design is effectively masked by the wide band noise of the high-side fan, and that the overall noise closely follows the slope of the ear response curve making for a more acceptable noise quality.

To reiterate, the best way to get rid of noise is to design it out. But where design for acoustical improvement must be secondary to appearance or performance, it is possible generally to reduce noise to acceptable limits by proper use of acoustical materials applied on the basis of logical reasoning processes.

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DISCUSSION

W. E. FONTAINE, West Lafayette, Ind.: What is the greatest noise-contributing factor in the machine, the noise created from gas flow or is it mechanical noise?

D. D. WILE, Los Angeles, Calif.: Quality of noise was mentioned. Can the author tell what method was used to evaluate quality, since that seems to be an individual or personal matter? Have the authors actually developed a jury, for instance? How extensively have they gone in getting the opinion of different people? What method was used to evaluate the individual reaction to the noise?

R. W. DOEG, Detroit, Mich.: The author mentioned a capacitor pick-up for *studying valve action*. Was it actually tried? If used, was it found that the valves were making several bounces during one cycle? Are other means of pick-up possible such as a strain gauge? Is the capacitor pick-up the ideal means?

C. M. ASHLEY, Syracuse, N. Y.: Because sound-pressure level readings are influenced both by the sound characteristics of the room in which the readings are taken and by

the location of the microphone with respect to the equipment being tested, no general significance can be attached to a specific set of readings. Thus, the evaluation of the raw data in terms of sones, or as a matter of fact in terms of any other criterion, tends to be misleading. Thus it is considered preferable either to present the raw data together with information concerning the room and the position of the microphone or to convert it to power level.

The formula of Stevens for converting octave-band sound-pressure readings to a Sone loudness scale is, I believe, generally recognized as the more accurate data by virtually everyone including Dr. Beranek.

There are several other factors which need to be considered besides loudness. These include problems of speech communication, annoyance and others which are related to the characteristics of the sound frequency spectrum as well as sound-pressure level. What is being dealt with here is an extremely complex subject.

R. S. BUCHANAN, Columbus, Ohio: Some of the illustrations show that the sound-level meter read in the range of 27 or 28 db. What part of the sound-level reading was the actual level of the room without equipment in it? In other words, what is the ambient noise level in the rooms?

L. C. BASTIAN, Philadelphia, Penna.: Do the authors feel that presently available techniques of sound measurement permit the preparation of a commercially acceptable method of testing and rating air-conditioning equipment? Do they believe that, in general, the octave-band method gives sufficiently reliable results or that it will be necessary in sound measurement of air-conditioning equipment to use one-third octave bands?

BRUCE LATTER, Louisville, Ky.: In the quality control type of examination using a sound room as shown in the paper, which method of obtaining data and what general criteria were used in determining whether or not a particular refrigerator is rejected or is acceptable? Have the authors been able to come up with an overall noise-level criterion drawing a definite line below which a refrigerator is acceptable and above which it is not?

R. S. BUCHANAN, Columbus, Ohio: By way of comment it would seem that the acceptance of any noise is a compromise. While it is interesting to talk about setting a level, it would seem doubtful if it is possible to set any level below which noise is acceptable, as this is a constantly changing conception. As equipment is made more quiet, people will expect it to be still more quiet. Acceptance of any noise is compromise, and the effort is to work toward a zero point.

AUTHORS' CLOSURE (Mr. Laughlin): Concerning the principal source of noise as mentioned by Mr. Fontaine, this can depend upon a particular compressor design. It has been observed that gas flow, motor forcing frequencies, and the basic pump mechanism all influence the noise of the compressor. Probably, the most variable noise in any one compressor design is mechanical noise since it can be influenced by so many factors. All the sources are considered in the noise evaluation.

The questions asked by Mr. Wile might well lead into quite a lengthy discussion. Briefly, the authors believe that noise quality is a subjective problem as it involves the question of how to assess customer acceptance of noise. Several approaches have been taken to try to evaluate noise quality. As one approach, a considerable number of refrigerators were tested in a typical residence and the results compared to corresponding tests in several sound rooms. Environmental influence differences were noted, as expected. However, the differences due to listener psychological reactions were found difficult to correlate. The attempt is to keep the noise level low enough so that it will be acceptable in the average home; indeed the quieter home. The aim is also to keep out any distracting tones which disturb the tranquility of the customers mind. Effort is directed at keeping out interruptions of the noise continuity; i.e., the

pulsating tone, since the ear is more disturbed by this type of noise. This is accomplished by the quality of the product being controlled throughout the entire assembly, from basic materials to final product, and noise tests made at the logical stages of assembly. There are two approaches in common use. The authors make use of field tests and sound juries. In the field test approach, the products are placed in people's homes and they are asked to report their reactions to the noise. The people include typical customers, as well as engineering personnel, in order to obtain varied subjective reactions.

A sound jury has also been utilized within the plant. It is hoped that the jurors are not influenced by their association with the product and are selected from other than engineering personnel for this very reason. It has been found that generally their impressions of the noise correlate with the control levels. This is very subjective, however, and no hard and fast answer is possible.

Mr. Doeg asked about the use of a capacitator pickup. A capacitor pickup was used; *i.e.*, a capacitor pickup was made out of the actual valve components. In some valve designs, a valve bounce was observed. This, however, was found to be controllable by the material and shape of the valve. Valve bounce can well be a source of noise, and hence the valves are evaluated under various dynamic loadings. It cannot be said that the capacitator pickup is the ideal way. The strain gauge undoubtably has a use although it has been found to be limited in its scope. When used, influences of the solubility of the oil-refrigerant mixture and the relatively high pressure-temperature relationships should be considered.

Concerning the remarks of Mr. Ashley, the authors hasten to state that one should take these levels as plotted advisedly, since they will vary, as Mr. Ashley pointed out, with the test environment.

Mr. Buchanan asked about the ambient noise level. The "A" network readings of the test rooms' background noise ranges from about 17 to 20 db. The better of the several rooms is 17 db, actually it is instrument noise except in the first and second octave bands.

It is believed that in his presentation, Mr. Ashley will cover the ground of Mr. Bastian's question, however, the authors' opinion is that the sound-power technique is the most appropriate method for such comparison. There is an inherent disadvantage using sound-pressure measurements (expressed either in decibels or sones) due to the environment influence. Rating methods based on data taken in sound pressure under other than anechoic or reverberant conditions must weight the environment influence.

The authors certainly believe that one should look at least at an octave band distribution of noise, and the one-third octave evaluation is by far the more informational. It is felt that it is not possible to express adequately all the indices of noise by any one value whether it be db, sones, phons, or watts. Hence, the need is implied for an expression for the influence of the noise by at least octave bands.

Tentatively, and along the general lines mentioned by Mr. Latter, there has been established a *rating criterion* for quality control based on tests made in a typical residence, the correlation being between the several sound rooms and actual use of jury evaluation of the noise in the residence and again in the sound room.

Although, this rating standard is based in part on the NC curves, modifications were made in order to consider the acoustic environment in which the testing is done, the impression of the jurors as to product quality, and the influence of air movement noise with its masking wide-band noise spectrum.

Thus there was established essentially a *go-no go* standard. This rating standard in the form of an octave-band curve is used in conjunction with a sone-rating level and an aural quality index. By endeavoring to obtain an *overall* index based on these three criteria it is felt that a reasonably good quality level is maintained.

1678

No. 1678

SOME METHODS FOR INVESTIGATING NOISE FROM COMPRESSORS USED ON HOUSEHOLD REFRIGERATORS

By R. C. BINDER*, PULLMAN, WASH.

VARIOUS questions may arise in planning a study of noise from refrigeration or similar equipment. This discussion concentrates on the investigation of certain machine noise problems. The methods reviewed are applicable to the study of the noise from small compressors of the type used on household refrigerators and are approached from the point of view of an engineer who desires to make a systematic study, with quantitative measurements and a minimum of trial and error.

NOISE

Any such study involves two different phases: (1) purely physical or engineering, (2) purely human. One of the simplest physical measurements to make is to use a sound-level meter. Sound from a machine acts on the microphone of the meter. This meter can be used to determine the overall sound-pressure-level, SPL, which is defined as

$$\text{SPL} = 20 \log_{10} \frac{P}{0.0002} \text{ decibels} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where P is the root-mean-square sound pressure, in dynes per sq cm, and 0.0002 is a reference pressure in the same units as P. The single reading of overall sound-pressure level, however, may not be sufficient. *Noise* is defined as *unwanted sound*. For example, two different sounds with the same overall SPL may provoke different

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human responses; one sound may be regarded as quiet or allowable, whereas the other sound may be regarded as *noise* by a human jury.

A much better method of measuring sound characteristics is to determine the sound frequency spectrum, or simply the sound spectrum, as illustrated in Fig. 1. The sound-pressure level, in decibels, is plotted as a function of frequency, in cps. Two different curves are necessary in a study of noise. In Fig. 1 the curve marked "machine" is determined by an engineering or physical measurement, with suitable instruments, of the machine itself.

The other curve in Fig. 1 marked *human* represents the sound-pressure level which a human jury evaluation regards as the limit of allowance or acceptance. The

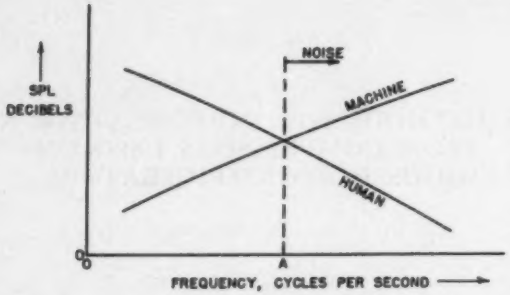

FIG. 1. . . .SOUND SPECTRUM FOR A MACHINE AND SOUND SPECTRUM FOR HUMAN ALLOWANCE

human allowance curve might be measured in various ways, as by a test of an established quiet machine. Sound-pressure levels above this allowance curve are objectionable, whereas levels below this allowance curve are acceptable. Consider frequencies below A; in this range the human allowance is higher than the machine sounds, therefore, the machine sound is not noise. Consider frequencies above A; in this range the human allowance is lower than the machine sounds, therefore, the machine sound is *noise*. Plots of the type illustrated in Fig. 1 can be used to specify noise in a definite scientific manner.

The human allowance curve may depend on a number of factors: the location of the machine; the purpose or activity in the particular location; the time of operation of the machine; and the people who might be listening to the machine.

Different criteria might be used to establish the human allowance curve. The different criteria might be divided into two general groups: one loudness and the other annoyance. Loudness and annoyance are not the same. There are many sounds that can be annoying, although not loud. For example, various intermittent sounds, such as a dripping faucet, can be annoying, although they are not truly loud. A loud sound is one at a relatively high sound-pressure level; a loud sound may be noise because it interferes with speech or it may cause damage to the ear. An annoying sound may be a low-level sound which has certain undesirable frequency characteristics.

A household refrigerator may be a low-level sound, yet be noisy because it is annoying. At the present time not enough data are available to establish just

what annoyance is. Parkinson and Jack[1] report that a variety of actual experiences in practice indicate that high frequency noise requires special attention, all out of proportion to its relative intensity. Some data presented by Parkinson[2] indicate that annoyance increases as the frequency increases. Some preliminary tests at Purdue indicate that annoyance is caused by high frequency components, above about 7,000 cps, and by discrete frequency components. For example, a low level metallic clicking, or intermittent action, may be annoying. Apparently the usual human response is to accept the more familiar low frequency sounds more readily than the less common high frequency sounds.

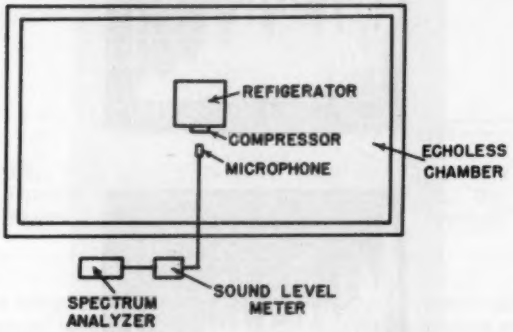

FIG. 2. . . .GENERAL ARRANGEMENT OF TEST FACILITY

TEST FACILITY ARRANGEMENT

Different test arrangements are possible. One general arrangement for a certain class of problems, as illustrated in Fig. 2, has various advantages. The compressor to be studied is connected to its load, and the entire refrigeration system is installed in an echoless or anechoic chamber. A microphone is placed close to the compressor. The signal from the microphone is led outside of the chamber to a sound-level meter; this meter gives a single reading of overall sound-pressure level. The signal from the sound meter is led into a spectrum analyzer, which gives a plot of sound-pressure level *vs* frequency, of the type shown in accompanying Fig. 1.

In an echoless chamber all the walls are so arranged that they absorb the incident sound, which comes from the machine. Glass fiber wedges provide a good absorbing material. Data on the design of echoless chambers can be found in a paper by Beranek and Sleeper.[3] An echoless chamber was built at Purdue with net inside dimensions of 16 x 12 x 7 ft. The glass fiber wedges were 24 in. long (perpendicular to the wall), and mounted on all interior surfaces. A wire mesh floor was installed above the bottom wedges. These wedges absorbed all frequencies above about 170 cps. This echoless chamber proved quite suitable for various studies of household refrigerators and other machines. The chamber provides a

[1] Exponent numerals refer to References.

controlled environment for doing detective work. A completely closed room isolates the testing area from other sources of noise, and provides a low background sound level. A microphone in the chamber picks up only the sound from the machine under test.

Suitable microphones and sound-level meters are available commercially. Various frequency analyzers are available commercially, as the octave-band analyzer,

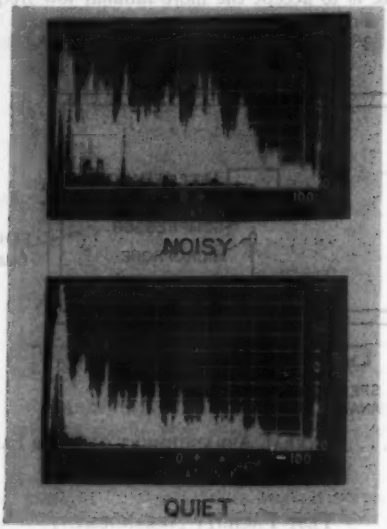

FIG. 3. . . .SPECTRUM ANALYZER TRACES TWO DIFFERENT REFRIGERATION COMPRESSORS

sound analyzer, and the spectrum analyzer. In considering a frequency instrument for low-level annoyance studies, it should be recognized that measurements of high frequencies may be quite important, as frequencies up to about 15,000 cps.

The spectrum analyzer[4] is one type of instrument that is particularly helpful in many noise studies. In this instrument, a cathode-ray tube gives the spectrum which can be photographed. An electron beam is swept horizontally about once a second. The horizontal sweep is so arranged that each horizontal position corresponds to a certain frequency. The applied sound pressure gives a vertical deflection to the beam. Fig. 3 shows the spectrum analyzer traces for two refrigeration compressors. In each trace the vertical scale is sound-pressure level, in decibels, whereas the horizontal scale is frequency, in cps, with the left end of the scale zero and the right end 20,000 cps.

In Fig. 3 the upper trace is for a noisy compressor, whereas the lower trace is for a quiet compressor. Each compressor had essentially the same low overall sound level. Each was designed for the same loading system. A comparison of the two

shows the difference between the two machines. The photographic negative for each trace can be used to make a comparison of the frequency spectra. Fig. 4 illustrates one type of comparison plot. For each frequency, the difference, in SPL, for the one compressor minus the SPL for the other compressor, is plotted.

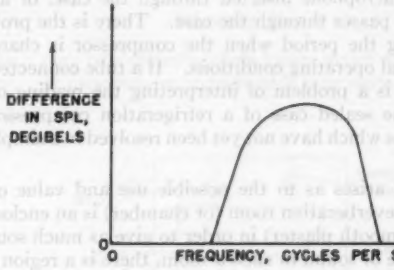

FIG. 4. . . .ILLUSTRATION OF PLOT IN WHICH THE DIFFERENCE IN SOUND-PRESSURE-LEVEL, SPL, BETWEEN TWO MACHINES IS PLOTTED AS A FUNCTION OF FREQUENCY

Such a plot brings out the difference between the compressors, and shows what has to be changed in order to get one machine to match another machine. The spectrum analyzer is notably convenient in giving an immediate picture of the sound spectrum. Compressor operation can be changed and the corresponding

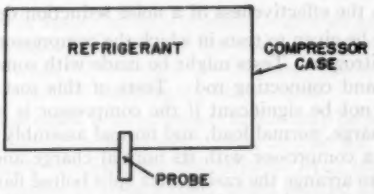

FIG. 5. . . .DIAGRAM OF PROBE ARRANGEMENT

changes noted immediately. Other types of analyzers may be used if other features are desired.

The foregoing discussion indicates a method for studying a noisy compressor, and indicating with quantitative measurements, just what results should be obtained in order to get a quiet compressor.

In some studies of noise location and detection, it would help if one or more sound probes could be inserted inside the sealed case. The general idea is illustrated diagrammatically in Fig. 5. If a sound measurement could be made inside the case, it could tell more about the sources of noise, and eliminate the question of

just what is transmitted through the case. If two microphones, some distance apart, could be inserted through the case and moved about, then a sound ranging technique might be used to identify sources. Possible sound ranging instrumentation, using the correlation of two different microphone signals, has been described by Gilbrech and Binder.[8]

One might consider a microphone inserted through the case, or a microphone connected to a tube which passes through the case. There is the problem of sealing the microphone during the period when the compressor is charged with refrigerant, and during normal operating conditions. If a tube connected to a microphone is used, then there is a problem of interpreting the reading of the microphone. Unfortunately, the sealed case of a refrigeration compressor presents a number of difficult problems which have not yet been resolved in a simple convenient manner.

Sometimes the question arises as to the possible use and value of a so-called *reverberation chamber*. A reverberation room (or chamber) is an enclosure in which the surfaces are hard (as smooth plaster) in order to give as much sound reflection as possible. With a source of sound in such a room, there is a region in which the sound is diffuse and the sound-pressure level is uniform. This makes it possible to arrange a convenient measurement of the sound power emitted by the sound source. The sound power is of value in certain studies, as in machine rating and in studies of absorption. In certain other studies, however, as those of location of sound sources and certain studies of noise reduction, the sound power may not be of real value. The reverberation chamber is useful for certain studies, and the echoless chamber is useful for certain other studies.

Arrangement of Compressor

Different types of compressor tests are possible. In some tests the objective may be to locate a major source or sources of sound. In some tests the objective may be to determine the effectiveness of a noise reduction device, as a muffler.

Consideration may be given to tests in which the compressor is operated with air, a vacuum, or with nitrogen. Tests might be made with some part removed, as a valve plate, piston, and connecting rod. Tests of this sort may be convenient, but the results may not be significant if the compressor is not operated with its normal refrigerant charge, normal load, and normal assembly of parts. It is most desirable to operate a compressor with its normal charge and load. One way of expediting testing is to arrange the casing with split bolted flanges so that the compressor can be dismantled and assembled easily.

Masking

In a machine there may be different noise sources, and there may be different paths from a single noise source to the human ear. In either case it is important to recognize the feature of "masking". Frequently, confusion results because this feature of masking is ignored. As an example, say there are two separate noise sources, one A and the other B in a machine. Say A is at a higher sound level than B. A masks or covers B. Reducing further the sound level of B will not weaken the sound level A which reaches the ear.

As another example, say there is only one noise source, but two separate sound paths, one A and the other B, from this source to the human ear. One path may

short circuit the sound; the situation is similar to that in an electric circuit. Say A is the better path (less resistance to noise transmission) whereas B is the poorer path (greater resistance to noise transmission). Reducing the transmission through B does not reduce the transmission through path A. A masks or covers B. If path A is weakened below B, then B can be heard.

POSSIBLE TESTS

In some cases it may be a difficult job to locate major noise sources in a refrigeration compressor. It may require considerable ingenuity to get an answer. Some

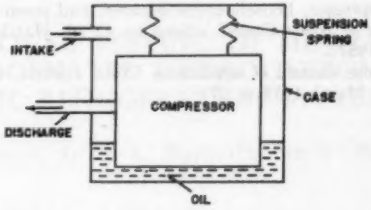

FIG. 6. . . .DIAGRAM OF COMPRESSOR UNIT

discussion of compressor, household-refrigerator and room air-conditioner noise has been presented by Webb[6] and Sawyer[7]. Some further discussion is presented in the following.

If foaming in the oil is suspected as a possible source of noise, various silicone anti-foam or defoaming agents are available for trial. Addition of a small quantity of a defoaming agent to the oil will cure the foaming. Thus a relatively simple test can be made to determine if foaming is a noise source.

In low-level annoyance cases, a valve impact action may be suspected as a source of noise. If it is too difficult, or otherwise unwise to cure the noise at the source, attention may be given to blocking the transmission from the source to the ear. This case illustrates a method of noise control which may be wise to follow in various other cases.

Fig. 6 illustrates diagrammatically a system with a reciprocating compressor. The noise can travel from the valve impact region through the compressor frame. This noise can pass through the metal suspension springs, the metal discharge pipe, and the oil bath to the outer case. From the case the noise travels through the air to the human ear. If the sound transmission through these three paths is blocked, then the listener will not hear the noise. A flexible tube can be placed in the discharge pipe to block transmission. Various elastomers and metal mesh mounting materials can be used in place of conventional steel springs to block transmission. Various means can be employed to isolate the oil bath from the case and thus block the paths. Tests with and without these modifications can be used to determine the effectiveness of the different noise reduction schemes.

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1679

No. 1679

ACOUSTICAL TESTING OF MUFFLERS FOR HALOGENATED HYDROCARBON AND OIL REFRIGERATION SYSTEMS

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THERE is a two-fold purpose for the muffler in a mechanical refrigeration system—to filter from the compressor discharge line those pulsations and noises introduced by the compression process and to serve as an accumulator for the compressor so as not to backload it. A rational design method based upon acoustical theory will yield a muffler which accomplishes both objectives. The method described herein is one which has been successfully used in the design of aircraft engine mufflers[1].

Since the muffler was designed by an application of acoustical theory it should be evaluated by acoustical methods to determine how well the theoretical design can be relied upon in practice. A component test of this type was therefore made. The test method utilizes a standing wave apparatus, loudspeaker, microphones, and the usual sound-level measuring equipment.

Muffler Design

A muffler configuration consisting of two expansion chambers connected by an internal tube was chosen because it would give high attenuation at the desired frequencies and could be incorporated into the rear head of the high-side rotary compressor. Schematically, the muffler is shown in Fig. 1. A qualitative explanation of its function can be made in the following manner.

A sound wave entering from the left will be partially reflected at the discontinuity at A. Of the sound energy which enters the first expansion chamber,

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** Research Laboratories, Whirlpool Corp.
† Research Laboratories, Whirlpool Corp. Associate Member of ASHRAE.
[1] D. D. Davis, Jr., George M. Stokes, Dewey Moore, and George L. Stevens, Jr.: Theoretical and Experimental Investigation of Mufflers with Comments on Engine—Exhaust Muffler Design (*NACA Report* No. 1192, 1954).

Presented at the Annual Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Lake Placid, N. Y., June 1959.

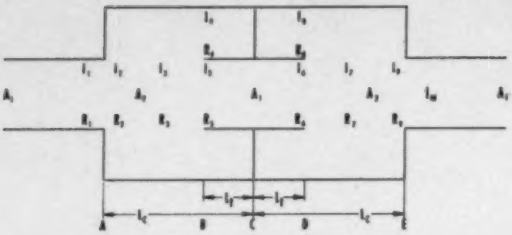

FIG. 1. . . .SCHEMATIC DIAGRAM OF SOUND DISTRIBUTION IN MUFFLER CONSISTING OF TWO EXPANSION CHAMBERS CONNECTED BY AN INTERNAL TUBE

some will form standing waves in it by reflection at the discontinuities at B and C and a portion will be transmitted through the connecting tube into the second expansion chamber. Standing waves will also exist in this chamber, and because reflections will occur at C and D, only a fraction of the sound energy in the second chamber will be transmitted into the exit tube.

The performance which can be expected of the muffler can be determined by the application of the transmission line equations developed in plane wave acoustical theory[1, 2].

The pressure existing in a plane wave distance x from the origin of the wave is

$$p = Ie^{j(\omega/c)(x-ct)} + Re^{-j(\omega/c)(x+ct)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

I = maximum pressure of the incident wave.
R = maximum pressure of the reflected wave.
$\omega = 2\pi f$.
c = velocity of sound.

In Fig. 1 the origin of x is at A, positive x to the right. The particle velocity is

$$u = \frac{1}{j\omega\rho}\frac{\delta p}{\delta x},$$

where

ρ = average density of the gas.

$$u = \frac{1}{\rho c}\left[Ie^{j(\omega/c)(x-ct)} - Re^{-j(\omega/c)(x+ct)}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The ratio of I_1 to I_{10} may be obtained by satisfying the boundary conditions at each point of discontinuity in the system by repeated application of Equations 1 and 2. At points A, B, D, E, the conditions to be satisfied are that the total pressure and volume flow are each continuous at a surface.

Thus, at A: $I_1 + R_1 = I_2 + R_2$ . . . . . (3)

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[2] P. M. Morse: *Vibration and Sound*, Chapter VI, (McGraw-Hill Book Co., Inc., 1948).

$$\frac{A_1}{\rho c}(I_1 - R_1) = \frac{A_2}{\rho c}(I_2 - R_2)(I_1 - R_1) = m(I_2 - R_2) \quad . \quad . \quad . \quad . \quad (4)$$

where

$$m = \frac{A_2}{A_1}$$

$$\text{At B: } I_3 + R_3 = I_4 + R_4 = I_5 + R_5 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$m(I_3 - R_3) = (I_5 - R_5) + (m - 1)(I_4 - R_4) \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$\text{At D: } I_6 + R_6 = I_7 + R_7 = I_8 + R_8 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$m(I_7 - R_7) = (m - 1)(I_8 - R_8) + (I_9 - R_9) \quad . \quad . \quad . \quad . \quad . \quad (8)$$

At E, assuming no reflected wave in the exit tube:

$$I_9 + R_9 = I_{10} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$m(I_9 - R_9) = I_{10} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

The boundary condition to be satisfied at C is that the particle velocity normal to the wall is zero.

Then

$$u = \frac{1}{\rho c}\left[I_4 e^{j(\omega/c)l_4} - R_4 e^{-j(\omega/c)l_4}\right] = 0$$

$$R_4 = I_4 e^{j2(\omega/c)l_4} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Similarly,

$$R_3 = I_3 e^{-j2(\omega/c)l_4} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

By using Equation 1, I_3 and R_3 are expressed in terms of I_2 and R_2, at $x = l_0 - l_1$. At B,

$$p = I_2 e^{j(\omega/c)(l_0 - l_2)} + R_2 e^{-j(\omega/c)(l_0 - l_2)} = I_3 + R_3 \quad . \quad . \quad . \quad . \quad . \quad (13)$$

Similarly at D,

$$p = I_6 e^{j2(\omega/c)l_2} + R_6 e^{-j2(\omega/c)l_2} = I_8 + R_8 \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

And at E,

$$p = I_7 e^{j(\omega/c)(l_0 - l_2)} + R_7 e^{-j(\omega/c)(l_0 - l_2)} = I_9 + R_9 \quad . \quad . \quad . \quad . \quad . \quad (15)$$

By solving these 15 equations simultaneously, the ratio of the incident sound pressure to the exit sound pressure is found to be:

$$\frac{I_1}{I_{10}} = \cos 2\frac{\omega}{c} l_0 - (m - 1)\sin 2\frac{\omega}{c} l_0 \tan\frac{\omega}{c} l_1 + \frac{j}{2}\left\{\left(m + \frac{1}{m}\right)\sin 2\frac{\omega}{c} l_0 + (m - 1)\tan\frac{\omega}{c} l_1\left[\left(m + \frac{1}{m}\right)\cos 2\frac{\omega}{c} l_0 - \left(m - \frac{1}{m}\right)\right]\right\} \quad . \quad . \quad (16)$$

The transmission loss of an acoustical device is defined as

$$10 \log_{10} \frac{W_1}{W_2}$$

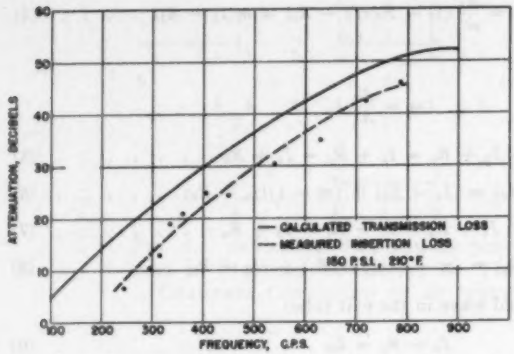

FIG. 2. . . .TRANSMISSION LOSS FOR A MUFFLER OPERATING IN A REFRIGERANT-OIL SYSTEM

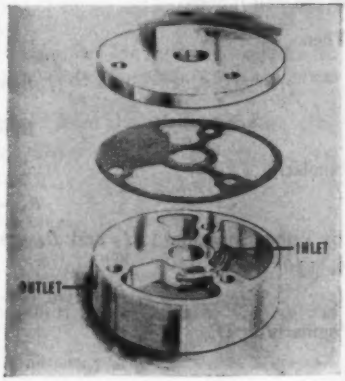

FIG. 3. . . .REAR HEAD MUFFLER WITH WARPED CAVITIES

FIG. 4. . . .ACOUSTICAL STANDING WAVE APPARATUS DESIGNED TO MEASURE TRANSMISSION AND INSERTION LOSSES OF THE MUFFLER

where

W_1 = intensity incident upon the device.
W_2 = intensity of the wave transmitted by it.

Since the intensity of a plane wave is proportional to the square of its pressure, the transmission loss for the muffler becomes

$$T.L. = 10 \log_{10} \left| \frac{I_1}{I_{10}} \right|^2 \text{ db} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

The squared magnitude of I_1/I_{10}, is found as

$$\left| \frac{I_1}{I_{10}} \right|^2 = \left[\text{Real} \left(\frac{I_1}{I_{10}} \right) \right]^2 + \left[\text{Imaginary} \left(\frac{I_1}{I_{10}} \right) \right]^2 \quad . \quad . \quad . \quad . \quad . \quad (18$$

The transmission loss calculated by means of Equations 16, 17, and 18 for a muffler to be operated in a Refrigerant 11-oil system at 150 psia and 210 F is shown in Fig. 2. The dimensions of the muffler are such that m, the area ratio, is 18. $l_e = 1\frac{1}{2}$ in. $l_1 = 1$ in.

These were chosen to provide appreciable transmission loss at frequencies less than 1000 cps.

The rear head muffler built to these dimensions is not a parallelepiped as is the prototype of Fig. 1, but rather the cavities are warped to fit the rear head, as shown in Fig. 3.

EXPERIMENTAL METHODS

An acoustical standing wave apparatus was designed to measure the transmission loss and insertion loss of the muffler. Its general design is similar to several described elsewhere[3]. Its features are evident in Fig. 4 and in Fig. 5.

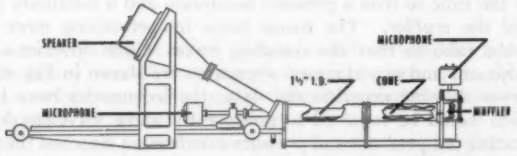

FIG. 5. . . .SCHEMATIC FOR THE ACOUSTICAL STANDING WAVE EQUIPMENT SHOWN IN FIG. 4

The output of a 6-in. loudspeaker is piped into the 1½-in. diam standing wave tube, and is transmitted from there into a conical tube, the apex of which is coupled to the muffler under test. In order to measure transmission loss, the sound pressure incident upon the muffler is required. This is measured by an encapsulated microphone to which is attached a small bore probe tube. The whole assembly is mounted on a carriage which slides along the bed to permit the measurement of standing wave maximum and minimum. Since the apparatus was designed to

[3] L. L. Beranek: *Acoustic Measurements* (John Wiley & Sons, 1949, pp. 326–329).

operate with the refrigerant at 150 psi and 210 F, the system is inclosed with a jacket for hot water.

The output end of the muffler is terminated in a section of steel tubing 20-in. long, whose cross section is the same as that of standard refrigerator tubing. This discharges into a chamber which is 40 in. long and 1½-in. diam. It is partially filled with glass wool in order to minimize standing waves in the output system.

This latter is the most crucial feature of the apparatus. In order to measure transmission loss it is necessary that the muffler be terminated in a reflection-free tube in which the sound pressure can be measured.

Because this has not as yet been realized in a sealed system under pressure as just described, measurements to date have been made with the system open to atmospheric air pressure.

The termination then consists of the same small bore tubing as is used in the sealed system, and over a 2-ft length of it is snugly fitted a 2-ft length of tube

FIG. 6. . . .EQUIPMENT MODIFICATION TO PERMIT ADJUSTMENT OF THE LENGTH OF THE TUBE SO THAT PRESSURE MAXIMUM AND MINIMUM CAN BE READ AT THE OUTPUT OF THE MUFFLER

which terminates in a 12-in. diam flange. The purpose here is to be able to adjust the length of the tube so that a pressure maximum and a minimum can be read at the output of the muffler. The flange helps by permitting more energy to be radiated by the tube so that the standing waves which do exist are minimized. The test equipment and experimental apparatus are shown in Fig. 6.

In the curves showing experimental data, the frequencies have been corrected to those which would be required to produce the same wavelength in the refrigerant at operating temperature and pressure as existed in air when the measurements were made.

TESTS AND RESULTS

Measurement of muffler insertion loss provides performance data which are representative of the actual attenuation provided by the muffler in the system in which it is measured. Insertion loss is defined as the difference in sound-pressure level, measured when the muffler is replaced by a tube of equal length, and when the muffler is inserted in the system, both measurements being made at a convenient point downstream from the muffler position. No elaborate mathematical formulation is required because the usual sound-level meter indicates sound-pressure level in decibels.

A microphone fitted with a short probe tube was used in making the insertion loss measurement. The open end of the microphone probe was inserted into the

steel tube which served as the output termination for the muffler. The end of the probe was flush with the wall of the output tube. In order to reduce the acoustical loading which the microphone and probe might place upon the system, the probe has a bore of only ¼-in. diam and is less than 2-in. long.

Tests were made to determine to what extent the microphone probe affects the system. Similar probes were fitted into the muffler terminating tube at 2 additional stations. Measurements of sound-pressure level were made both when the additional probes were inserted and when they were replaced with solid rods. The differences in sound-pressure level were barely detectable on the meter.

The results of the insertion loss measurement are shown in Fig. 2, with the calculated transmission loss. The lowest frequency at which measurements were

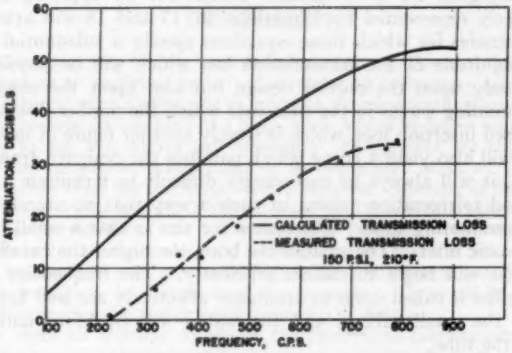

FIG. 7. . . .EXPERIMENTALLY DETERMINED AND CALCULATED TRANSMISSION LOSSES COMPARED

made is the lowest frequency at which the output tube propagates a wave outward. The highest frequency was dictated by the ambient noise level. It is seen that the measured insertion loss closely parallels the transmission loss calculated on the basis of no reflections in the output system.

The measured insertion loss, however, is a function of the ideal transmission loss and the reflections which exist in the output tube. The magnitude of the attenuation in a refrigerator installation, however, will depend upon the configuration of the tubing which connects the muffler to the succeeding component in the system.

Transmission loss data are especially meaningful if the measured transmission loss is

$$10 \log_{10} \left| \frac{I_1}{I_{10}} \right|^2$$

under the condition that there is no reflected wave, for that is the basis upon which Equation 18 is developed. Even though the muffler was terminated in a tube in which reflected waves did exist, it was possible to measure I_{10} as well as I_1. The transmission loss calculated therefrom, however, will not be the same as that

predicted for the muffler, because the magnitude of I_{10} is affected by the presence of a reflected wave in the output tube.

Determinations of I_1 and I_{10} were made at discrete frequencies from measurements of the maximum and minimum sound-pressure levels which existed in both the input tube and output tube. The tedious calculations outlined in the Appendix were required to extract the magnitude of the incident waves, I, from the maxima and minima. The experimentally determined transmission loss using Refrigerant 11 is plotted in Fig. 7. Again, the calculated transmission loss is included for comparison.

CONCLUSION

A muffler designed for a refrigerator compressor by applying the acoustical theory succinctly represented by Equations 16, 17 and 18 will attenuate in the range of frequencies for which those equations specify a substantial transmission loss. The magnitude of the transmission loss which will be physically realized depends not only upon the muffler design but also upon the existence or nonexistence of standing waves in the tube into which the muffler discharges.

The measured insertion loss, which is simply another figure of merit for muffler performance will also yield a curve which parallels the designed transmission loss.

In practice, it will always be exceedingly difficult to terminate a muffler with commonly used refrigeration tubing in such a way that no standing waves will exist in the terminating tube. The reason for this is that a small bore tube is a high pass acoustic filter. The smaller the bore, the higher the cutoff frequency at which the tube will begin to radiate efficiently. The frequencies of the waves which the muffler is called upon to attenuate effectively are well below the cutoff frequency of the small tubing, and the waves will therefore undergo multiple reflections in the tube.

APPENDIX

In a standing wave, the ratio of the magnitude of the reflected to the incident pressure is given as[4]

$$\left|\frac{R}{I}\right| = e^{-2\pi\alpha}$$

The general expression for the magnitude of the pressure in a standing wave is

$$|P| = 2Ie^{-\pi\alpha}\sqrt{\cosh^2(\pi\alpha) - \cosh^2(\pi\beta)}$$

Consequently,

$$|P_{max}| = 2Ie^{-\pi\alpha}\cosh(\pi\alpha)$$

$$|P_{min}| = 2Ie^{-\pi\alpha}\sqrt{\cosh^2(\pi\alpha) - 1} = 2Ie^{-\pi\alpha}\sinh(\pi\alpha)$$

These are the quantities measured indirectly in the input and output tubes. $|I|^2$ is separated from $|P_{max}|$ and $|P_{min}|$ by taking the product:

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[4] P. M. Morse: *Vibration and Sound*, Chapter VI (McGraw Book Co., Inc., New York, 1948).

$$P_{max} | \cdot | P_{min} | = 4P e^{-2\pi\alpha} \cosh(\pi\alpha) \sinh(\pi\alpha)$$

$$P = \frac{| P_{max} | \cdot | P_{min} |}{4e^{-2\pi\alpha} \cosh(\pi\alpha) \sinh(\pi\alpha)} = \frac{| P_{max} | \cdot | P_{min} |}{2e^{-2\pi\alpha} \sinh(2\pi\alpha)}$$

The hyperbolic angle is found from the relation

$$\left| \frac{P_{min}}{P_{max}} \right| = \tanh(\pi\alpha)$$

The maximum and minimum values of sound-pressure level measured by the sound-level meter are converted to $| P_{max} |$ and $| P_{min} |$ by applying the definition

$$SPL = 20 \log_{10} \frac{P}{P_{ref}}$$

DISCUSSION

C. M. ASHLEY, Syracuse, N. Y.: Too often a problem of the sort described by Mr. Martz is handled on an empirical basis. It is refreshing to have an approach starting from a rational theory, then attempting to correlate the experimental work with theory. If more work were done on this kind of a basis, there would be faster progress.

I would like to ask Mr. Martz two questions. First, what are the primary frequencies from the compressor which need to be removed? Most of the compressor noise normally arises from the discharge valve pulsation. Because of the sharpness of the pressure front and also because of the repeated oscillations following the opening of the valve, higher harmonics of the valve discharge frequency tend to predominate. For instance, for a 2-cylinder compressor running at 3500 rpm the most objectionable noise may not be the 116 cycles corresponding to the fundamental pulsation frequency but may be the 3rd, 4th or 5th harmonic of this frequency. It would be very desirable to know what frequencies the author tried to absorb or isolate.

Second, was there an attempt to adjust the theoretical calculations to take into account the actual terminal impedance of the set-up in which the tests were run?

Was an effort made to adjust the muffler design to fit the system terminal impedance?

As a further comment, it would seem that a calculation of the terminal impedance should not be too difficult. The impedance is a function of the pipe size and the end conditions at the point where the pipes enter into the chambers on the compressor and condenser. While the problem of calculating of terminal impedance may seem difficult, it certainly is not by any means out of the question.

It must also be recognized that if there were no absorption or radiation from the ends of the system, there would be no attenuation in the muffler. The determination of the amount of radiation from the system is quite important.

AUTHORS' CLOSURE (Mr. Martz): The frequency of the compressor was 114 cycles, and the harmonic of the greatest amplitude was 342 cycles per second. This is the harmonic which we are most interested in attenuating.

No attempt was made to adjust the conditions as mentioned by Mr. Ashley. It was thought that it would be simpler to account for the termination by measuring the sound pressure maxima and minima at the outlet of the muffler. It was felt that it would be too difficult to adjust the design because we did not have a precise formulation of the driving point impedance of the terminal tube.

The authors agree with Mr. Ashley's other comments, and greatly appreciate his encouragement.

C. E. PLOEGER, Evansville, Ind.: Would this approach allow one to correct for the pumping capacity and power loss in the system usually caused by addition of a muffler to the compressor?

MR. MARTZ: Yes, to the extent that this method does not cause a restriction in the system. Forecasting probably is not permissible, but it can be said that the method does not backload the system. So, whatever pumping capacity was present to begin with should not be greatly influenced by this muffler.

MR. PLOEGER: The statement does not agree with some of the results with which this discusser has been familiar. One of the results attempted by his group is to get a quiet muffler and to resolve the power and capacity losses which go with the muffler. There does not seem to be any correlation between these qualities in the muffler design method under discussion.

MR. MARTZ: This muffler has not been completely evaluated as installed on a compressor. It was designed, however to have a flow area such that no restriction will be introduced. What was described here was a "cold" test, one in which there was no fluid flow.

1680

No. 1680

SUPPRESSION OF OSCILLATIONS IN GAS-FIRED RESIDENTIAL† HEATING EQUIPMENT

By C. F. SPEICH* AND A. A. PUTNAM**, COLUMBUS, OHIO

This paper is the result of research sponsored jointly by the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, the American Gas Association and the Oil-Heat Institute of America, Inc., conducted through Battelle Memorial Institute

OSCILLATIONS, as differentiated from vibrations and other noise sources‡, have concerned manufacturers, installers, and users of residential heating equipment for many years. Through experience gained by trial and error, a number of suppression techniques have been found which are effective in some, but not all, circumstances. Lack of understanding of the causes of oscillations forestalled understanding the reasons for the uncertainty of success of the suppression techniques. The entire oscillation problem has been further aggravated in recent years by the use of smaller, more compact heating units placed nearer to the living quarters of residences. Also, people in general have become more noise conscious. Hence, in the light of these developments, a research program jointly sponsored by the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, the American Gas Association, and the Oil-Heat Institute of America, Inc., was initiated at Battelle Memorial Institute in May of 1954, to study the fundamental causes and methods of suppressing oscillations and pulsations in both gas- and oil-fired residential heating equipment.

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† This paper is essentially similar to the paper published under the same title and appearing in the ASHRAE JOURNAL, August 1959, p. 41. The paper is an abridgment of an original report on the cooperative project. For information concerning the unabridged report entitled, The Suppression of Oscillations in Gas-Fired Residential Heating Equipment, write to ASHRAE Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio. See also, Oscillations in Gas-Fired Heating Equipment and Their Suppression, A. G. A. Par Plan Report, Project No. DG-7-Gu, November 1959.

* Principal Mechanical Engineer, Battelle Memorial Institute.

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‡ Other noise sources might include random flame noise, panel vibrations, vibrations of internal elements due to vortex shedding from sharp edges, start-up and shut-down noises, ignition spark amplifications, and the like. This paper, however, will deal only with combustion-driven oscillations.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N.Y., June 1959.

The original of this paper is the fifth in a series of reports[1,2,3,4] on the results of the research program. Techniques for the suppression of oscillations in gas-fired heating equipment utilizing burners of both the single- and multiple-port type are first outlined in simplified form for ready reference. Since the mechanisms of oscillation for each type of burner unit differ, the remainder of the paper has been divided into two sections, one devoted to each type of burner. Each section includes a brief summary of the mechanism of oscillation, and outlines in limited detail the techniques that can be used to suppress the oscillations.

OUTLINE OF SUPPRESSION PROCEDURES

The sciences of combustion physics and heat transfer are intimately connected with the design of gas-fired heating equipment. When oscillations occur in these units, it must be realized that a third science, namely acoustics, also enters into the problem. In the past, when objectionable oscillations occurred, frequent resort was made to change some of the variables affecting the combustion process in an attempt to suppress the oscillations. Unknowingly, these changes also affected the acoustic behavior of the heating unit. However, since these acoustic characteristics were unknown, the trial-and-error methods of suppressing oscillations were not always effective on a consistent basis.

This paper is intended to present information on the acoustic aspects of the combustion oscillation problem. Therefore, it is believed that designers, installers, and service representatives should all find information which will be helpful in accomplishing the suppression or elimination of oscillations in gas-fired heating equipment. It is realized that not all of the methods suggested will be practical from the standpoint of the combustion and heat transfer specifications which must be met for a particular unit. Therefore, compromises will have to be made. The intent in this paper, however, is not to identify what methods of oscillation suppression are best for one particular unit, but rather, to outline the general methods which could be used in designing or modifying a large number of different units.

To make maximum use of the suppression techniques outlined herein, it is necessary to know how near the normal furnace operating conditions are to those operating conditions for which the maximum amplitudes of oscillation occur. For single-port units, it will generally be sufficient to know whether the existing oscillating frequency is less than or greater than the resonant frequency of the furnace.

For multiple-port units, one might picture a relief map of intermixed flat-topped hills and valleys where the hills represent oscillating regions and the valleys, non-oscillating regions. The normal operating conditions of a particular unit will place it somewhere on the hypothetical map. If at these operating conditions, the unit is found to be on one of the hills and thus found to oscillate, then it is obvious that ways must be found to get down into a valley. It is apparent, however, that the relative locations of these hills and valleys dictate the appropriate means of reaching a valley. As is shown in the original version of this paper, the most practical means of finding the location of these hills and valleys is to conduct experimental tests on a particular arrangement of burner face at the port level and to determine how extensive a change in one or several variables is required to stay out of oscillation.

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[1] Exponent numerals refer to References.

TABLE 1*a*. . . .FACTORS AFFECTING OSCILLATION AMPLITUDE FOR SINGLE-PORT BURNER UNITS

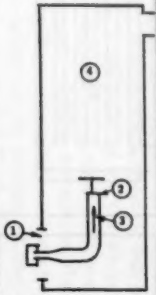
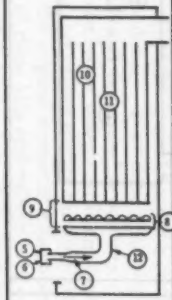
	Furnace Characteristic or Adjustment	Effect of Change in Furnace Characteristic or Adjustment on Amplitude of Oscillation		Time When Suggested Procedure is Applicable	Remarks and Applicable References
		Observed Frequency			
		Less Than Resonant	Greater Than Resonant		
Single-Port Burner Units					
	Secondary-air opening (1)	Increasing either item lowers amplitudes of oscillation	Decreasing either item lowers amplitudes of oscillation	Design, installation, service	Effect on resonant frequency slight; increases acoustic radiation to outside; effect on overall noise output unknown because of nonlinear effects of oscillation (see Appendix C, Ref 4)
	Burner-port diam (2)			Design, possible future application in installation & service	Prevention of flashback and blowoff require changes in flow rate and mixture ratio
	Primary-air flow rate (3)	Decreasing either item lowers amplitudes of oscillation	Increasing either item lowers amplitudes of oscillation	Design, installation, service	Decreasing flow rate may cause CO, CO_2, and stability problems; increasing flow rate will cause increases in random combustion noise and instability of flame
	Volume of combustion chamber (4)			Design	Only practical for proposed designs
	Firing rate	These items have little or no effect on amplitude of oscillation			Except at high primary-air flow rates
	Distribution of secondary air				
	Secondary-air flow rate				
	Length of burner tube				Increase in length produces slightly lower amplitudes of oscillation
	Temperature of combustion gases				Indeterminate effect on the amplitude of oscillation
	Burner position				Indeterminate effect on the amplitude of oscillation
	Spreader height				
	Spreader diameter				

TABLE 1*b*. . . .FACTORS AFFECTING OSCILLATION AMPLITUDE FOR MULTIPLE-PORT BURNER UNITS

	Furnace Characteristic or Adjustment	Effect of Change in Furnace Characteristic or Adjustment on Amplitude of Oscillation: Observed Frequency	Time When Suggested Procedure is Applicable	Remarks and Applicable References
	Multiple-Port Burner Unit			
	Firing rate (5)	Increases in these variables will increase chances of oscillation	Design, installation, service	The chances of increasing or decreasing oscillations is based on the average of a large number of units. For a particular unit, the relation of the normal operating conditions to those conditions where oscillations occur may cause the effects of various furnace characteristics to contradict this table. For additional material, see Reference 3.
	Primary-air flow rate (6)		Design, installation, service	
	Primary-mixture ratio (7)		Design, installation, service	
	Altitude (firing-rate constant)		Installation	
	Ratio of flame length to dark space (8)	Increases in these variables will decrease chances of oscillation	Design, installation, service	
	Distance from burner port level to heat exchange(9)		Design; only small changes possible during installation & service	
	Damping losses (10)		Design	
	Frequency of heating unit (11)		Design	
	Frequency of burner (12)	Effects of oscillations average out	Design	

Tables 1*a* and 1*b* present for both single- and multiple-port burner units a summary of all of those variables which have been found (1) to affect the oscillation amplitude or (2) to have little or no effect on the amplitude. For these tables, it is assumed that when one variable is changed, all others are held constant. As can be seen in the table for single-port-burner units, the effect of changing the variables will depend on whether the observed frequency of oscillation is less than or greater than the resonant frequency of the heating unit. It should be possible to ascertain the relative value of the observed frequency with respect to the resonant frequency by noticing what effect each variable would have on the amplitude of oscillation.

For the multiple-port burner units, it should be emphasized that the effects from changes in the various variables hold on the average for a large number of units. Any particular unit may react in just the opposite manner. Thus, the need for knowing where the operating conditions are with respect to those of the oscillating regions is made doubly important.

For a more complete understanding of the suppression techniques described in the table, a list of applicable references is given in the right-hand column. These references are the significant publications resulting from this overall research program. The remaining sections of this paper review briefly the essential features of the research results that have been presented in considerable detail in the referenced publications.

REDUCING OSCILLATIONS IN SINGLE-PORT BURNER UNITS

The Mechanism of Oscillation: The results of the preliminary investigations and a detailed discussion of the proposed mechanism of oscillation of single-port burners have been presented in two of the previous papers of this series[1, 4]. In essence, the mechanism which has been proposed states that there are 2 phenomena involved in the generation of the oscillations as follows:

1. An aerodynamic vortex-shedding from the lip of the burner port;
2. A periodic pressure oscillation in the combustion chamber, with a frequency close to the resonant frequency of the chamber.

The vortex-shedding produces a periodic variation in the rate of heat release of the flame. It is the coupling between the rate of heat release and the pressure oscillation which determines the amplitude, and to a small extent, the frequency of the oscillation. The difference between the vortex-shedding frequency and the frequency of the pressure oscillation determines the extent of this coupling. If this frequency is small, the oscillation will lock in together, only one frequency of oscillation will occur, and the amplitude of oscillation will be high. If, on the other hand, this difference in frequency is large, the amplitude will be small especially if the frequency of vortex-shedding is the lesser of the two frequencies.

Fig. 1 shows the correlation obtained using data from one burner of constant port diameter external to the furnace. In this correlation, the Strouhal number is plotted against the Reynold's number. Strouhal number, often used in studies of vortex-shedding phenomena, is defined as the product of the vortex-shedding frequency and the burner-port diameter divided by the average velocity of the combustible gases through the burner port. The Reynold's number, defined as the ratio of the inertial forces to the viscous forces of the gases is commonly written as the product of the average velocity of the combustible gases through the burner port and the burner-port diameter divided by the kinematic viscosity of the gases.

The fact that the correlation between the Strouhal and Reynold's numbers was good when the burner was external to the furnace indicates that vortex shedding is of an aerodynamic origin rather than a result of some acoustical effect related to the combined burner-furnace space configuration. Thus, the normal frequency of the vortex shedding is dependent only on the design geometry and flow conditions at the burner lip.

Techniques for Suppressing Oscillations: The suppression of the oscillations in single-port burner heating units depends on the ability to decrease the coupling effects between the rate of heat release and the pressure oscillation. Basically,

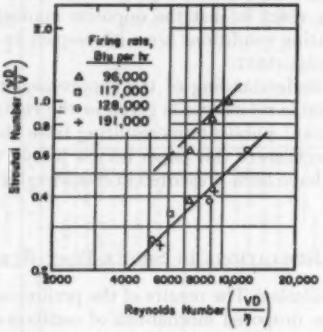


FIG. 1....STROUHAL NUMBER AS A FUNCTION OF REYNOLDS NUMBER FOR THE SINGLE-PORT BURNER EXTERNAL TO THE EXPERIMENTAL FURNACE

this can be accomplished by increasing the difference in frequency between the two phenomena.

Consider first the method of suppression by changing the average port velocity. Using the data from Fig. 1 and assuming that the kinematic viscosity of the gases and the burner-port diameter are constant, it can be shown that the frequency of vortex-shedding varies directly with the average velocity of the gases passing through the burner port. If, for example, a heating unit is considered to produce objectionable oscillations when the frequency of vortex-shedding is less than the resonant frequency of the heating unit, then increases in the average port velocity will cause an increase in the amplitude of oscillation. These increases in oscillation amplitude will continue until the frequency of vortex shedding exceeds the resonant frequency of the unit. Further increases in the vortex-shedding frequency will cause a general decrease in the amplitude of oscillation. As a general rule, it can be said that oscillation suppression can be achieved by making changes in the average port velocity so as to cause the difference between the vortex-shedding and the resonant frequencies to increase.

Next, examine the method of suppression by changing the burner-port diameter. Since each of the dimensionless ratios used in the correlation depicted by Fig. 1 have as one term the burner port diameter, it would be expected that changes in

this diameter would affect the frequency of the vortex shedding. However, the 1:1 slope of the lower line drawn through the data on Fig. 1 means that when the two terms are equated, the diameter term cancels out. Since it is volume rate of flow of the combustible gases issuing from the burner which is of major concern to burner design, this volume rate of flow can be substituted for the average burner-port velocity when equating the two dimensionless terms. This substitution re-enters the burner-port-diameter term, taken to the fourth power, into the resulting equation. With the equation in this form, and the volume rate of flow kept constant, it is found that the frequency of vortex shedding will vary inversely as the fourth power of the burner-port diameter. It should be realized that changes in burner-port diameter will also change the average velocity of the gases leaving the burner port. If the diameter is increased, both the frequency and flow velocity will decrease. However, as a complication, the flashback and blow-off limits will both be changed by these modifications in burner-port diameter and flow velocity. It may be necessary to change the volume flow rate or the mixture ratio, or both in order to attain satisfactory combustion conditions.

As part of this investigation, several experimental burners were built which had port diameters different from that of the single-port burner generally used in these studies. These studies did show that oscillation suppression could result from diameter changes.

As a third method of suppression, consider altering vortex-shedding characteristics. Changes in the burner lip that alter the type of vortices shed were also expected to alter the amplitude of oscillation. For instance, a burner with a spiral lip of the proper advance might cause the vortices to shed continuously. Other configurations might cause the vortices to shed irregularly, thus preventing a periodic heat release. The breaking up of the large vortices into smaller ones is still another possible means of reducing the amplitude of oscillation. Small vortices not only tend to decay faster than larger ones, but tend to cancel each other. This last type of configuration was found to be very effective in reducing the amplitude of oscillation.

As a fourth method of suppression the effect of furnace volume changes can be thought of. As previously indicated, the amplitude of the pressure oscillation is related to the closeness of the natural vortex-shedding frequency to the resonant frequency of the heating unit. Thus, reductions in amplitude should be possible not only by changing the frequency of the vortex shedding away from the furnace frequency, but also by changing the resonant frequency of the heating unit away from the vortex-shedding frequency. Since the resonant frequency of the heating unit is a function of its volume and internal configuration, decreasing the volume will increase the resonant frequency of the unit. Thus, changes in unit volume can be used as a means of increasing the difference in the two frequencies.

REDUCING OSCILLATIONS IN MULTIPLE-PORT BURNER UNITS

The Mechanism of Oscillation: Multiple-port, ribbon, or slot burners differ from single-port burners in the manner in which the oscillations are driven. The difference is that vortex shedding is not involved. Rather, the periodic heat-release rate necessary for driving results from a pulsation of the supply of the mixture of fuel gas and primary air to the flame. This means by which combustion oscillations may occur has been long recognized, but only recently, in two previous papers of

this series[1, 2], has it been associated with the multiple-port type of burner used in residential heating units.

There are three furnace-burner variables which determine the presence of oscillations in multiple-port burner heating units. These are:

1. Natural frequency of the burner.
2. Natural frequency of the heating unit.
3. Length of time, called the time lag, needed for the combustible gases to pass from the burner-port level to an average region of combustion in the flame.

Using a mathematical treatment, such as shown in Reference 3, it is possible to relate these variables to predict regions of oscillation alternating with regions of non-oscillation. On the basis of the most simplified mathematical analysis, there are an infinite number of these regions, but in practice the number is limited to a few only.

Fig. 2*a* indicates this type of array for the simple theory. The symbols used are defined as follows:

ν = natural frequency of heating unit.
τ = time lag.
ν_b = natural frequency of burner.
θ = measure of the ratio of flame height to the distance of the flame base above the burner port level.

The button-shaped flame is a theoretical concept which depicts the flame as being infinitesimally thin and flat. Deviations from this flame shape to one of the form of a wedge or cone will alter the regions in which oscillations can occur. An example is shown by the bar graphs to the right of the figure.

When acoustic damping occurs, the regions of oscillation are also decreased as shown in Fig. 2*b*. Damping occurs from friction in the system or acoustic radiation from the system and represents a means of taking energy out of the oscillating system.

Suggested Means for Determining the Oscillating Regions in Multiple-Port Burner Units: Fig. 2 is not the most advantageous array to use to determine the oscillating regions of a particular unit. This arises because of the difficulty of calculating accurate values of time lag. As defined in Reference 3, the time lag is a function of, among other factors, the flame speed and dark space for the fuel-air mixtures used in the combustion process. Not enough data on these items are available in the literature to permit computation of accurate values of the time lag. Therefore, a procedure has been devised by which the problem of the calculation of time lag has been circumvented. The aim of this procedure is to produce a set of curves of the changeover lines corresponding to the horizontal lines of Fig. 2, but in terms of firing rate, mixture ratio, and observed frequency of oscillation. The ratio of observed frequency to burner frequency would determine which of the alternating regions would be prone to cause oscillation.

In this procedure, it is suggested that a series of investigations be made using one design of burner-port level and a number of different sized burner bodies and different lengths of heat exchanger sections in order to obtain a range of burner and heat exchanger natural frequencies. It should be emphasized that the frequency is the important factor and exchangers built up of available components or flat plates are sufficient.

Two techniques are outlined in the original version of this paper by which the data may be analyzed. In the first, the mixture ratio is held constant and the firing rate varied. As the firing rate is increased in small steps, the ranges of oscillation and corresponding frequencies are recorded. The reciprocal of the frequency is plotted against the reciprocal of the primary gas-flow rate, for each set of data.

In the second method of handling the data, attention is focused on one small increment of frequency, given by the fundamental mode of one heat exchanger and

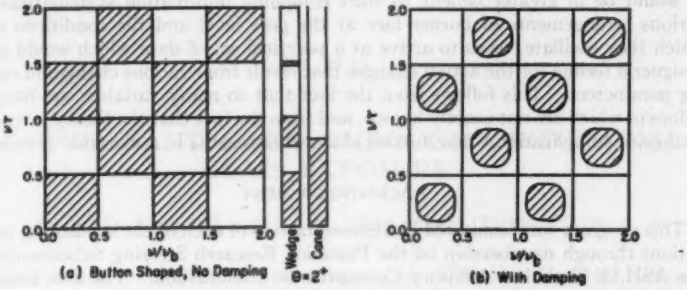


FIG. 2. . . .REGIONS OF PULSATION IN SIMPLE MULTIPLE-PORT BURNER

possibly higher modes of larger exchangers. Firing rate is held constant for each run, and mixture ratio is varied. Whether or not oscillations occur at a series of closely spaced points is determined and indicated on a plot of percent theoretical primary air as a function of volume flow rate of primary mixture.

These methods will establish the locations of regions of oscillation and non-oscillation in terms of easily measured furnace and burner variables. Using this information as a basis estimates can then be made as to the vulnerability to oscillation of a new heating unit design with this same arrangement of burner face at the port level. In the case of other units where oscillation problems exist, estimates can be made of the necessary changes required to produce an oscillation-free unit.

CONCLUDING REMARKS

With most research programs, the answering of one set of questions often leads to the asking of a host of others. This has been true in this program. It is believed that there are at least 2 items related to single-port burners brought to light in this program which require further study. One of these is the physical mechanism by which the vortices are shed from the burner port lip. At the same time, the variation of the vortex-shedding frequency with burner-port diameter, configuration, and flow velocity should be studied more extensively than was done during this program. This information would be valuable to the designer of burners, not only in determining more completely what designs would have the least chance of producing objectionable oscillation, but also in leading to a general understanding of rates of heat transfer and mixing associated with such burners even when combustion-driven oscillations are not present.

A second program of interest for single-port burner heating units would be to study in more detail the coupling effect between the periodic variation in heat release and the pressure oscillations. Included in this investigation should be an intensive study of the feedback effect which is hypothesized as being the triggering mechanism needed to sustain the vortex shedding. It is believed that if this coupling effect were better understood, more suppression techniques might well be brought to light.

It would appear that, from other than an academic standpoint, further basic research on multiple-port units would not be particularly profitable at present. It would be of greater benefit to start compiling information systematically on various arrangements of burner face at the port level and the conditions under which they oscillate, so as to arrive at a practical set of data which would give a designer a feeling for the actual changes that result from various changes in operating parameters. This follows from the fact that so many variables are involved, values of which are not exactly known, and from the fact that the theory is reaching a stage of complication where further elaboration results in a marginal return.

ACKNOWLEDGMENT

This program was monitored by representatives of each of the sponsoring organizations through membership on the Pulsation Research Steering Subcommittee of the ASHAE Technical Advisory Committee on Combustion. The able assistance of the members of that subcommittee in guiding the work, and in providing necessary information and materials is gratefully acknowledged. Special thanks are due to personnel of the Armstrong Furnace Co. and the Surface Combustion Corp., both of Columbus Ohio, who provided the experimental furnaces and burners used in this investigation. It is also desired to acknowledge the assistance of W. R. Dennis and Gail M. Clough in obtaining various portions of the data.

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1681

No. 1681

SOLAR RADIANT GAINS THROUGH DIRECTIONAL GLASS EXPOSURE

By R. D. CRAMER* AND L. W. NEUBAUER**, DAVIS, CALIF.

THIS IS a description of some of the recent work of Western Regional Housing Project W-8 and University of California research project 1536. Reports on previous work concerned with temperature studies in an experimental house trailer and in existing inhabited houses have been published[1–6]. The trailer has small window openings, simulating a conventional house in which orientation is not critically important because the windows are distributed about evenly on all 4 sides and because radiant gains through the limited glass areas are comparatively low. Subsequently, an 8-ft cubicle was acquired to make available a structure in which orientation has a pronounced effect on interior air temperatures. The results obtained with its use form the basis for this paper.

THE CUBICLE

This unit was designed to represent a room or small house with a large proportion of its wall area given over to glass. It is 8 ft square and 8 ft high. It has a floor of ¼-in. plywood on 2 × 4-in. joists, without insulation.

Beneath is a low *crawl space* open to the outside air on all sides. Three of its walls are framed with 2 × 4-in. studs, insulated between with 2-in. mineral wool, and sheathed on both interior and exterior surfaces with ¼-in. plywood. They contain no windows. The flat roof-ceiling has the same construction.

The fourth wall is glass, 8 ft square, made of ⅛-in. sheet glass, simply framed. The framing is painted white, as are the other exterior wall surfaces and roof. The interior finish is natural DF plywood.

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** Associate Professor of Agricultural Engineering, University of California.

[1–6] Exponent numerals refer to References.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N. Y., June 1959. Published in ASHAE JOURNAL SECTION, Heating, Piping & Air Conditioning, November 1958, p. 155.

Fig. 1 shows the glass wall of the cubicle and gives some indication of the flat open terrain of its locale. The cubicle is portable and can be moved with a crane or large fork lift. However, for purposes of the tests described herein (except as indicated) one location only was used.

It should be noted that while the glass muntins are small in section, they do cast some shadow when taken in aggregate. A wall made of an uninterrupted

FIG. 1. . . . THE CUBICLE (LOOKING NNE). SHOWING GLASS AND SOLID WALLS

sheet of glass might produce somewhat higher interior air temperatures than those recorded in Fig. 2 but probably the difference would be slight.

INSTRUMENTATION

The principal installation for obtaining temperatures in the cubicle consists of 12 thermocouples of copper and constantan wires located in the surfaces of the walls, roof, ceiling, and floor, as well as in the air indoors and out. Temperature readings were taken manually, every hour during the day, and occasionally at night, using a self-compensating standard electrical potentiometer (reading di-

rectly in F deg) and a 12-point switch. This equipment was kept and operated in one corner of the cubicle. As the hourly temperatures of roof, wall, floor, and air were read they were recorded by hand. Mercury thermometers and a recording hygrothermograph were used to verify thermocouple readings.

CLIMATIC CONDITIONS

These tests were conducted by the Experiment Station at the University of California at Davis located approximately in the center of California's Central Valley, latitude about 38½ deg N.

In this region the average daily maximum temperature in July and August exceeds 95 F. Temperatures above 90 F occur on 23 days of each month. Readings exceed 100 F on 9 days in July and on 7 days in August. Natural cooling usually takes place at night, however, as the average minimum for July and August in Davis is 54 F. Minimum temperatures fall below 60 F on 27 nights in July and on 28 nights in August. These figures are based on 40-year averages (1910–50) and were obtained from the University Weather Station located just a few hundred feet from the cubicle. University Weather Station data were also used to obtain the reported outside air temperatures.

In Davis the high mid-day temperatures are accompanied by low humidities, thus rendering the heat relatively less objectionable. But according to the 1956 HEATING, VENTILATING, AIR CONDITIONING GUIDE, the temperature zone found most comfortable in summer for people at rest, ranges from 71 F at 100 percent RH (relative humidity), to 76 F at 50 percent RH and to 82 F at 10 percent RH. These conditions are exceeded in the region during the heat of day for the full summer period.

Rain during the summer is virtually unknown. Even clouds and/or daytime winds are infrequent. The usual day is hot, dry, clear and still, followed by evening cooling breezes from the south. These conditions recur with a regularity surprising to one accustomed to eastern or midwestern climates.

MEASUREMENTS

The cubicle was placed in an open field on plowed earth entirely unshaded except for low buildings and trees at some distance to the east and west. These produced partial shade during parts of the first and last hours of sunshine, but their effects were not substantial.

During June, July, and August 1956, temperature readings of the exterior and interior surfaces of the cubicle and of the inside and outside air were taken and recorded hourly from early morning until late evening. Data collected for the infrequent days which were windy or cloudy or cool were recorded but not used in the averages.

Each curve in the figures in the paper represents average temperatures for a period of not fewer than 3 typical days. Temperature readings were plotted using the outside air dry-bulb as a base. In each case the hourly readings of the average outside air temperatures for a particular 3- or 4-day run were subtracted from the hourly readings of average inside air temperatures for that run. The differences were then added to the average Davis summer air temperature curve for the recent 5-year period (1951–55) and so plotted. This method makes possible

a direct comparison of the cubicle's performance during the several runs, regardless of differing outside air temperatures.

It should be noted that the 5-year curve is lower (cooler) than the older 40-year average. This difference partially can be explained by the fact that the 5-year

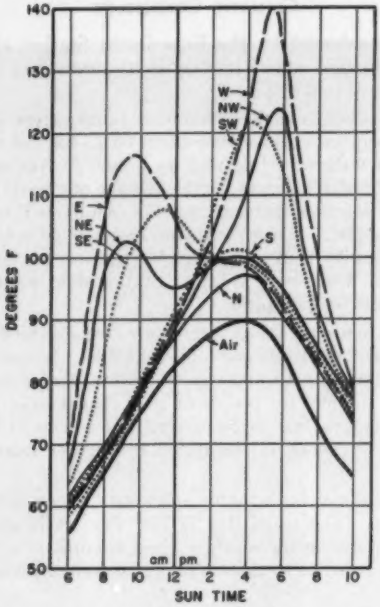


FIG. 2. . . .EFFECTS OF ORIENTATION. SHOWING THE BEHAVIOR OF THE INSIDE AIR OR ROOM TEMPERATURE AT DIFFERENT GLASS ORIENTATIONS AS COMPARED WITH AN AVERAGE SUMMER OUTSIDE AIR OR WEATHER STATION TEMPERATURE CURVE. THE SYMBOLS REPRESENT CARDINAL AND INTERMEDIATE POINTS OF THE COMPASS

average is for the entire summer while the 40-year average covers the 2 hottest months only. However, the more recent 5-year data does indicate somewhat lower temperatures. Whether this signifies a short- or a long-range trend is as yet unknown.

Climate is continually variable. No 2 days are exactly alike nor is there any such thing as perfect clearness, stillness, or dryness. Idiosyncrasies of individual days inevitably introduce irregularities in experimental results. In Davis during

the summer these idiosyncrasies are represented to some degree by haziness and/or wind during the day, and by nocturnal southerly breezes which vary in magnitude. But the methods reported here seem reliable because several opportunities to verify earlier results have produced consistent measurements.

PLOTTING

In general, the figures shown in this paper represent the results of each experiment plotted first against the 5-year average summer outdoor air previously described, and then against the average outdoor air curve flattened out and represented as a straight line or zero reading. The former method shows the total thermal condition; the latter subtracts out convected exchange with the outdoor air and confines itself to radiant effects against the wood surfaces and through the glass.

In the attempt to include a number of curves on each graph to facilitate multiple comparisons, some difficulty in reading closely adjacent lines was encountered. To alleviate this condition, lines which actually overlapped and which in some cases crossed back and forth were straightened out and plotted as close parallels. Corrections used to make this possible never exceed 2 deg and seldom exceeded 1. The hours involved tend to fall during morning and evening periods where comfort is not a serious problem.

LIMITATIONS

In examining the results, certain operating characteristics of the cubicle should be kept in mind. First, heat sources are made up of conducted gains through the roof, floor, and walls, and of radiant gains through the glass. The cubicle is kept tightly closed and infiltration is negligible.

Another important consideration is that the cubicle is not ventilated. Therefore, interior air temperatures indicate the magnitude of the problem which ventilation and/or mechanical cooling encounter in their efforts to produce conditions of comfort. For this reason interior air temperatures exceed exterior air temperatures during most of the day, resulting in conducted heat losses through the glass wall, through the floor to the relatively cool crawl space air below, and to some minor degree (for the most part negligible) through the north wall when cardinal orientations are being used. For this reason, also, the cooling of the cubicle in the evening shows some lag which largely would be alleviated if it were ventilated generously.

The cubicle's shape deserves comment. In comparison with 1-story houses, it has proportionately greater wall surface and smaller roof surface. Since roof gains usually exceed wall gains this should indicate that a rectangular solid more nearly approaching the shape of an average small house would produce interior air temperatures at least as high if not somewhat in excess of those measured here.

ORIENTATION

The cubicle was rotated at intervals to face the glass wall in 8 different directions. In addition to the 4 cardinal points, north, south, east, and west, facings of northeast, southeast, southwest, and northwest were tested. Substantial differences

were observed in interior temperatures as these changes were made, as indicated on the graphs of Figs. 2, 3, and 4.

In Fig. 2 the air curve represents the 1951–55 summer average air temperature previously described. Since the south readings were taken at the summer solstice the south curve represents the lowest (coolest) one which would be experienced during the summer. Its height may be expected to increase progressively during July and August.

On June 21 in Davis, the morning sun is due east at about 8:15 a.m. suntime. It begins to enter the cubicle, when the glass is facing south, at 8:30 and stays

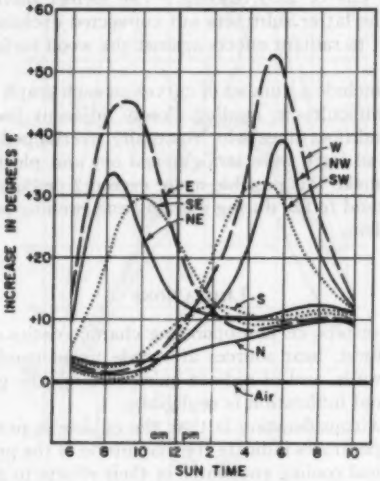


FIG. 3. . . . EFFECTS OF ORIENTATION. SHOWING THE DIFFERENCES BETWEEN THE ROOM AND WEATHER STATION TEMPERATURES AT DIFFERENT GLASS ORIENTATIONS

until 3:30. This means that it shines through the glass for only 7 hr, at a steep vertical angle (75 deg at noon). The result is an interior air temperature curve for the morning hours only slightly higher than that produced when the glass faced westerly, and received no sun. The glass facing north produced the highest early morning temperatures, because some radiation is admitted prior to 8:00 a.m.

During the late afternoon (after 4 p.m.) glass orientations from north through east to south are shaded. They cool uniformly with only the slight effects of lag differentiating between them.

One might expect the northeast curve to rise earlier than the east curve, and conversely might expect the northwest curve to drop later than that representing west. In the former case, the runs were made early in July, somewhat after the

solstice, and so the sun rose about 25 deg north of east and was shaded by horizon buildings and trees until it reached a point about 20 deg north of east. By 6 a.m., when recordings began, it was about 15 deg north of east and thus more nearly normal to east-facing glass than to northeast. In the evening the cooling lag requires more time to reduce the very high west peak than the lower northwest peak. The lag more than compensates for the fact that the latter occurs about ½ hr later.

The times that the peaks occur result from combinations of the influences of radiation and convection, and of the delays caused by the insulating qualities of

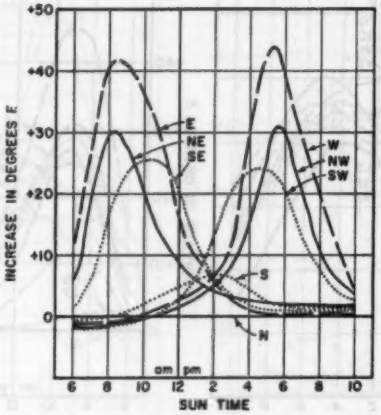


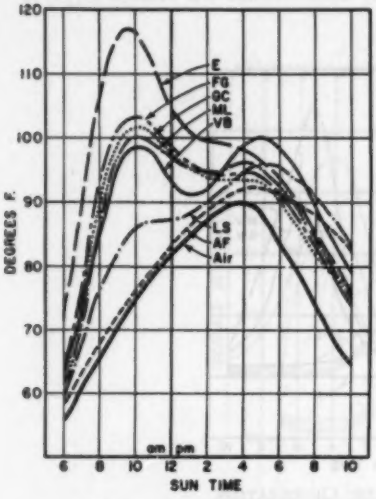
FIG. 4.... EFFECTS OF ORIENTATION. SHOWING THE DIFFERENCES BETWEEN ROOM TEMPERATURES WHEN THE GLASS WAS FACING NORTH AND ROOM TEMPERATURES WHEN IT FACED EACH OF 7 OTHER DIRECTIONS

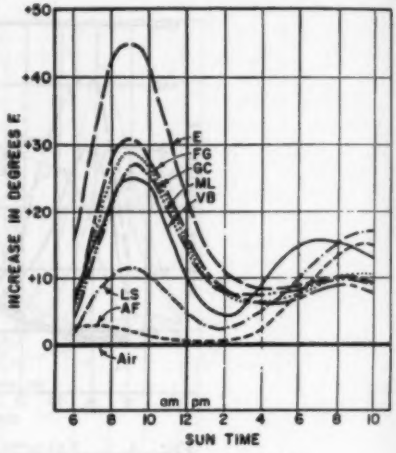
the construction and the specific heat of the materials. Since quantities of direct radiation which penetrate the glass when it is facing north are small, the extent to which the north curve exceeds the outside air temperature can be attributed essentially to gains conducted through the roof and through east, west, and south walls. Roof and wall temperatures during exposure to the sun (and for a cooling period thereafter) exceed air temperatures. The heat conducted through their mass is made up of radiation gains to exterior surfaces minus some surface cooling by convection.

Fig. 3 plots the outside air temperature as a flat line, or zero reading. By subtracting the air temperature from all readings, the influence of convected gain is eliminated leaving curves which are made up of radiation gains: partly against opaque surfaces stimulating conduction through walls and roof, and partly directly

through the glass wall. Again the north curve shows just the former with a roof surface and 3 walls involved. This curve could be expected to be higher were these surfaces painted black instead of white. The time distances between noon and the peaks are now the results of delays of insulation and specific heat only. The combined effect of these influences would appear to be about 2 hr.

The reverse shape of the westerly curves from 7 to 10 a.m. derives from the maximum rate of outside air temperature rise during those hours. The apparent





E = uncovered glass (east).
FG = glass fiber boucle drapes.
GC = glass cloth curtains.
ML = milium lined drapes.
VB = metal venetian blinds.
LS = linen roller shade.
AF = aluminum foil.

FIG. 5 (LEFT) AND FIG. 6. (RIGHT). . . . EFFECTS OF SCREENING DEVICES. SHOWING ATTEMPTS TO REDUCE THE IMPACT OF RADIATION GAIN ON CUBICLE ROOM TEMPERATURES USING VARIOUS MATERIALS MOUNTED INSIDE EAST-FACING GLASS

peak in the easterly curves at 8 p.m. is caused by the tapering off of the rate of decrease of the outside air at that time. Again it should be noted that with evening ventilation the right hand portion of all curves could be lowered.

Fig. 4 shows the same glass orientations, this time plotted against a flattened-out, zero-reading north curve. Since the north curve represented radiation influences on conduction through solid walls, this item is now subtracted out along with the convected influence of the outside air previously eliminated in Fig. 3. The remaining Fig. 4 curves represent gains resulting solely from radiation through the glass wall.

The symmetry of the easterly and westerly curves is marked and seems to straddle an axis located at about 1:30 p.m. This would indicate a cooling delay,

for this portion of the heat gain alone, of something like 1½ hr since the sun is symmetrical about the noon line. The peak of the south curve seems to reinforce this conclusion.

The variations in the height of the easterly and westerly curves average out evenly (east *vs.* west) suggesting that their slight differences can be attributed to

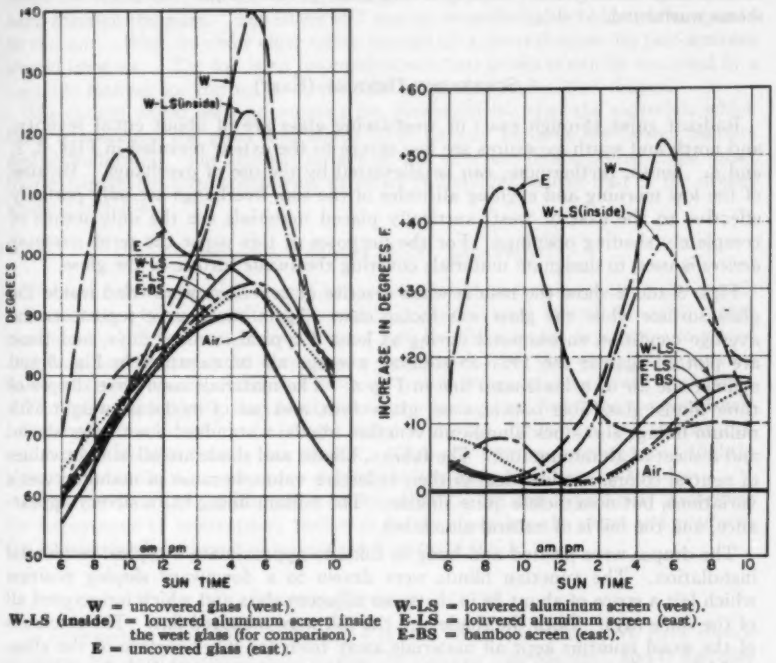


FIG. 7 (LEFT) AND FIG. 8 (RIGHT). . . .EFFECTS OF SHADING DEVICES. SHOWING ATTEMPTS TO REDUCE THE IMPACT OF RADIATION GAIN ON CUBICLE ROOM TEMPERATURES USING VARIOUS MATERIALS MOUNTED OUTSIDE EAST-FACING AND WEST-FACING GLASS

the climatic idiosyncrasies of the particular days of the individual runs. The consistently thicker easterly curves, however, can perhaps be explained by the fact that easterly orientations reach their inside air temperature peaks early and are attempting to cool during the period of the day when the rise of the outside air temperature is at its maximum rate.

The westerly and south curves fall slightly below the north curve at dawn but remain slightly higher than north during the evening. This difference can probably be attributed to the cooling effects of nocturnal southerly breezes from which the north glass orientation only is completely sheltered.

Since radiation gains through easterly facing glass result from rays which begin in a horizontal position and end in a vertical position and since the reverse is true of solar incidence on west-facing glass, it might be expected that the shapes of the curves for these 2 conditions would vary with the temporal distribution of radiation quantities, even though total quantities for east and west are the same. Apparently, however, in spite of these differences in geometric relationships, the conclusion that easterly and westerly-facing glass produce equal radiant heat gains seems warranted.

SCREENING DEVICES (EAST)

Radiant gains through east- or west-facing glass are of about equal severity, and north and south exposures are less severe to the extent revealed in Figs. 2, 3, and 4. South, furthermore, can be alleviated by the use of overhangs. Because of the low morning and evening altitudes of the sun, overhangs are only partially effective to the east or west; vertically placed materials are the only means of completely shading openings. For the purposes of this paper the term *screening devices* is used to designate materials covering the *inside* surface of the glass.

Figs. 5 and 6 show the results when a series of coverings were tried inside the glass surface while the glass was facing east. Again each curve represents the average condition encountered during at least 3 typical summer days, and these are plotted against the 1951–55 summer average air temperature in Fig. 5 and against the air as a horizontal line in Fig. 6. The materials used were drapes of three kinds: glass fiber boucle, sheer glass cloth, and one of moderate weight with milium lining; also stock aluminum venetian blinds; a standard linen roller shade; and a sheet of aluminum foil. The fabrics, blinds, and shade are all of light values of neutral colors, not identical in their reflective values because of manufacturer's variations, but nonetheless quite similar. The milium lining has a silvery appearance, and the foil is of natural aluminum.

The drapes were pleated and hung in folds to approximate a typical residential installation. The venetian blinds were drawn to a downward sloping position which left a space of about ¼ in. between adjacent slats and which intercepted all of the sun's rays. Both the shade and the foil were mounted flat. The thickness of the wood muntins kept all materials away from the inner surface of the glass, and left an air space between the glass and the covering material of about 3 in.

All covering materials were left mounted around the clock rather than being removed during the night. For this reason the cooling of the cubicle in the evening was retarded by virtue of the insulating effect which the covering material added to the glass. (The largest single source of heat loss of the cubicle is by conduction through the glass to the outside air.) This condition produces temperature curves which are high during evening hours and reflects physical differences among the various covering materials. For example, venetian blinds produced the highest evening temperatures probably because of their greater specific heat. They act as a *radiator* in the conventional sense in dissipating their sensible heat to the cubicle air mass. The linen shade and aluminum foil tend also to retard evening cooling, perhaps because of their relative imperviousness to air filtration as compared with the more porous and loose-fitting drapes. This latter explanation may also be the reason for the higher evening position of the drape with milium lining as compared with the unlined glass fiber and sheer types. The slight variations in relative

position of the venetian blinds at dawn and the sheer curtain at dusk can probably be attributed to climatic idiosyncrasy.

The top curve represents the glass facing east with no protection. The next 3 show the 3 drapes which cluster consistently close together so that it is doubtful whether any distinction can be made among them. The venetian blind curve is slightly lower while exposed to the sun but has the disadvantage of slower cooling. Both the shade and the foil do a reasonably complete job of reflecting radiation back through the glass. The shade still retains an undesirable bulge while exposed to the sun. Were its color clear white instead of a neutral *oyster* its performance should improve. The foil is an impractical solution unless it can be mounted in a movable manner for control as in the case of blinds, shades, and drapes.

The results would seem to warrant the generalization that the materials which hang flat against the glass and which have smooth surfaces are better barriers against radiation. The smooth surface of the venetian blinds seems to be offset by the corrugated form comprising its series of slats, not unlike that of the drapes hanging in folds.

SHADING DEVICES (EAST AND WEST)

The term *shading devices* is used here to designate materials covering the *outside* surface of the glass. Results of runs where materials were mounted outside rather than inside the glass are given in Figs. 7 and 8. On the east exposure a louvered aluminum screen and a bamboo screen were used. On the west the aluminum louvered screen was repeated and was used inside the glass as well for comparison.

The morning bulge in the aluminum screen facing east indicates the small percentage of radiation which the louvers failed to shade. By comparison the bamboo was completely effective in this respect. Thus the bamboo curve stayed slightly below the aluminum screen curve during the cooling period later in the day.

A comparison of the 2 aluminum screens facing west (inside and out) underlines the importance of intercepting radiation before it strikes the glass and reaffirms the relative ineffectiveness of a *corrugated* material inside.

SHADING DEVICES (SOUTH)

In conjunction with Figs. 2, 3, and 4 it was noted that the south data shown represent the most favorable time during the summer (the solstice), and it was also noted that further improvements are possible with shading devices.

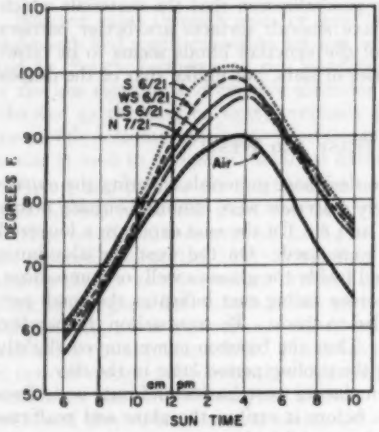
Figs. 9 and 10 show the cubicle air temperature during the basic south exposure with the glass unprotected during the solstice compared: (1) with the outside air temperature; (2) with the use of aluminum louvered screen and ordinary wire screen outside the glass; and (3) with the glass facing north.

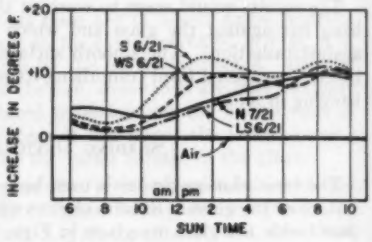
Ordinary wire screen accomplishes some small degree of shading at high solar altitudes but its effectiveness would doubtless decrease later in the summer with a lower sun. One might expect it to be effective hardly at all if used on an east or west exposure. On the other hand the curve representing the aluminum louvered screen can be expected to remain much as it is as the summer wears on because of the extra shading *capacity* provided by the configuration of the louvers. They are designed to keep out all summer sun. A comparison of Figs. 7, 8, 9 and 10 might suggest that aluminum louvered screen is relatively more useful east and west than on the south. Late August or early September readings, however, would probably reveal a more effective south performance than do the June readings.

It is interesting to note that glass facing south, fully shaded (as it is with the use of aluminum louvered screen) produces a somewhat cooler interior than uncovered glass facing north (assuming the south wall to be opaque and unshaded).

The explanation derives from the fact that in the former case cubicle radiant gain comes from a roof and 2 walls (east and west) while in the latter case it comes from a roof and 3 walls (east, west, and south).

Again, as in Figs. 7 and 8, the louvered aluminum screen curve, in relation to the others, is slightly low at dawn and slightly high during the evening. An explanation for the former may lie in the slope of the individual louvers exposing





At summer solstice:

S 6/21 = uncovered glass.
WS 6/21 = ordinary wire screen.
LS 6/21 = aluminum louvered screen.

One month after solstice:

N 7/21 = north facing glass for comparison.

FIG. 9 (LEFT) AND FIG. 10 (RIGHT). . . . EFFECTS OF SHADING DEVICES. SHOWING ATTEMPTS TO REDUCE THE IMPACT OF RADIATION GAIN ON CUBICLE ROOM TEMPERATURES USING VARIOUS MATERIALS MOUNTED OUTSIDE SOUTH-FACING GLASS

them rather well to the nocturnal sky for radiant cooling. The latter situation is probably caused by the shielding of the glass from the south cooling breezes.

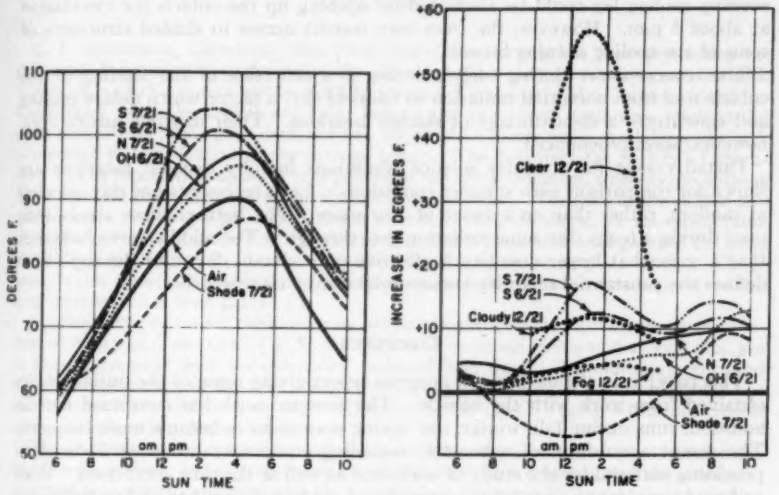
SEASONAL VARIATION AND SUMMER SHADE

Another series of runs compared curves for unprotected south glass during June and July and the winter solstice, with overhang and tree shade on south glass, and with north facing glass.

The highest summer curve represents an unshaded south glass exposure one month after the solstice (during the latter part of July). This might be considered an average summer condition since it falls somewhere near the middle of the hot weather and represents an average solar path. It is interesting to note in both Figs. 11 and 12 that the July peak temperature falls earlier in the afternoon than the solstice peak which again is earlier than the north peak. These differences are caused by the fact that the north gain is virtually all conducted gain through

opaque walls initiated by the radiant gains of their surfaces, while the 2 higher south curves represent progressively proportionately larger quantities of direct radiant gain through glass, producing higher inside air temperatures, with less delay and causing more rapid cooling by conduction through the glass.

Performance of a solid plywood overhang which kept out all sun during the middle of the day was also observed, and it can be seen by comparing Figs. 9 and



At summer solstice:

S 6/21 = unshaded glass.
OH 6/21 = glass under an overhang.

One month after solstice:

S 7/21 = unshaded glass.
N 7/21 = unshaded glass facing north for comparison.
Shade 7/21 = cubicle under complete tree shade.

FIG. 11 (LEFT) AND FIG. 12 (RIGHT). . . . EFFECTS OF SHADE. SHOWING ATTEMPTS TO REDUCE THE IMPACT OF RADIATION GAIN ON CUBICLE ROOM TEMPERATURES USING VARIOUS MATERIALS OUTSIDE SOUTH-FACING GLASS. IN FIG. 12 THREE INDIVIDUAL WINTER SOLSTICE DAYS LABELED CLEAR, CLOUDY, AND FOG ARE ADDED FOR COMPARISON

10 with 11 and 12 that it is about as effective as an aluminum louvered screen; slightly more effective at the peak but slightly less effective during evening cooling. The differences are small and so the overhang and the screen should be considered similar. The slight evening upturn of the overhang curve might be explained by the shelter it gives from the cold night sky and/or by the fact that relatively warm air pockets may be denied their gravitational rise and be trapped under the overhang and against the glass. Again it should be noted that shaded south glass gives better performance than does the glass facing north.

The curve representing the fully shaded condition gives an average of days when the cubicle was moved under a grove of trees with lawn covering the ground under and around it. The trees provided complete shade against the sun's rays except during brief and negligible periods at sunrise and sunset. At these times some radiation filtered through to cast spots of sunshine on the east and west walls, but for all practical purposes shade was complete.

The results produce comfortable temperatures well below the outside air as measured by the weather station thermometer in a nearby exposed location. The evening cooling lag could be alleviated by opening up the cubicle for ventilation at about 5 p.m. However, the trees may restrict access to shaded structures of some of the cooling evening breezes.

The reverse curve during early morning is a reflection of the *shading* of the cubicle roof from nocturnal radiation to the cold sky, a factor which delays cooling and constitutes a disadvantage of shaded locations. Their daytime advantages, however, seem pronounced.

Partial curves for 3 winter solstice conditions in fully exposed locations are shown for comparison with summer conditions. Each represents one day selected at random, rather than an average of 3 or more. The bottom curve shows that even during a foggy day some radiation gets through. The middle curve indicates that a somewhat larger quantity is effective on a cloudy day, and the top curve defines the substantial quantity encountered during clear weather.

COMMENTS

This paper is in the nature of a progress report giving some of the initial results obtained from work with the cubicle. The summer work has continued and in addition, runs during fall, winter, and spring seasons have become more frequent. The recent acquisition of automatic recording equipment produces 24-hr data providing material for the study of nocturnal as well as daytime conditions. Wall and roof temperatures are being recorded and examined together with calculations comparing them with theoretically expected readings. Further structural and environmental variables are under examination.

It should be noted that these experiments are being performed in climatic conditions typical to much of the Southwest where radiation values and air temperatures are high and humidities low. Under these circumstances environmental conditions which provide shade have relatively greater effect than they would have in cloudier and more humid climates. The results given here cannot be applied in such areas without interpretation.

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DISCUSSION

L. F. SCHUTRUM, Cleveland, Ohio (WRITTEN): The Society has been conducting experiments for some time to determine the solar heat gains through windows covered with draperies, and this report will be published soon. Fortunately Mr. Cramer visited the Laboratory when the project was being started and as a result, tests were made in the solar calorimeter of the glass cloth drapery which Mr. Cramer made available and which is reported in Figs. 5 and 6 of the authors' paper. The data from the Society's Laboratory will be reported in terms of Btu gain from the sun's radiation; whereas the authors have reported their results in terms of air temperature rise. The ratio of the cubicle air-temperature rise for the glass cloth drape to that of the glass alone, for the peak value reported in Fig. 6, it about 0.6. The ratio of instantaneous solar heat gains of the glass cloth drapery to that of glass alone, as determined from measurements made in Cleveland, is also about 0.6. This shows somewhat of a relation between air temperature rise and instantaneous heat gains.

Similarly, the ratios for the aluminum louvered screen; inside on the west-facing window at 5:00 p.m.—sun time, Fig. 8; and the screen on inside facing east at 9:00 p.m. are in good agreement with the calculated ratios of heat gain. The aluminum louvered screen on the outside of the west-oriented window, differs from the estimated results but this is most likely due to some slight difference in the profile angles, which are critical when the solar altitudes are low.

The instantaneous hourly heat gains due to solar altitudes were calculated for the window glass alone, and in combination with the glass cloth drapery, and the aluminum louvered screen, and compared with the temperature rise curve for an east-facing window as shown in Figs. 6 and 8. The instantaneous heat gain and the temperature rise curves are very similar, with the instantaneous curve reaching a maximum earlier in the morning. This indicates that the air temperature rise in the cubicle is somewhat delayed by thermal storage. When the sun is in the west, the instantaneous heat gains through an east-facing window due to the solar and sky radiation, are very low. However, the curves in Fig. 8 for the east-facing window show an increase in air temperature rise which indicates that appreciable heat gain in the late afternoon must be through the other 5 surfaces of the cubicle.

HARRY BUCHBERG, Los Angeles, Calif.: In listening to the presentation one point which was not clear was how convection was separated out from the curves.

A second question is whether any attempt was made to determine the reflectance of the materials tested. It seems that reflectance might serve as an index for determining the effectiveness of the various materials. There should be a direct relationship between the test results and the solar reflectance of the materials tested.

C. D. POTTER, Providence, R. I.: Was there any experiment with sun-reflecting glass to kill radiant heat? Something of that nature is now on a job where the owner insists it reflects the sun heat. It is a colored glass. Were there any experiments with that?

J. W. ZAHRADNICK, Amherst, Mass.: Did the cubicle have a sloped roof or a flat roof?

AUTHORS' CLOSURE (Mr. Cramer): Convected gain only was separated out. In Fig. 2 the outside air temperature for the average condition was plotted. In Fig. 3 the air was

plotted as a flat line, or zero reading. In other words differentials were plotted. The results were presented as differences between outside temperature and inside temperature which have been essentially the result of radiant gains minus convected losses.

Reflectances were not determined. Mr. Schutrum of the Laboratory staff has indicated in a letter that he subsequently determined the reflectances.

In answer to the question of glass, no glasses other than the ordinary window glass were tested. Other work has been done on other kinds of glass.

Although the photo in the paper may produce an illusion of a sloped roof, actually the cubicle roof was perfectly flat.

1682

No. 1682

LOAD CALCULATIONS USING PRETABULATED ADMITTANCE FUNCTIONS†

By W. B. DRAKE*, SUNNYVALE, CALIF., HARRY BUCHBERG**, AND D. LEBELL‡, LOS ANGELES, CALIF.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS in cooperation with the Department of Engineering, University of California, Los Angeles.

BEFORE the widespread use of modern computers, Mackey and Wright[1, 2, 3] developed methods for the computation of periodic heat transfer at the inside surface of separate homogeneous wall sections. These computational procedures are based on the analytical solution of the differential equation of heat conduction and involve certain approximations. Johnson[4] extended this work presenting temperature amplitude decrement and time lag values in terms of generalized property and boundary conductance moduli for homogeneous walls.

Equivalent temperature differentials for calculating heat gains through structural sections are tabulated in THE GUIDE††. These data are based on the Mackey and Wright solutions and computed for a fixed value of surface conductance and surface absorptivity. Stewart[5], in setting up the equivalent temperature differential method, pointed out that some judgments and arbitrary adjustments were made as a necessary expedient. In addition, the temperature differentials are tied to single sol-air temperature functions and corrections that are not adequate for many applications.

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† This paper is based on one part of an investigation carried out in partial satisfaction of the requirements for the degree of Master of Science in Engineering.

† This paper and the following paper, Chapter No. 1683, were both discussed at the same time. See discussion on page 540.

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[1] Exponent numerals refer to References.

†† HEATING, VENTILATING, AIR CONDITIONING GUIDE 1957 (published by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS).

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N. Y., June 1959.

Greater economy and precision in engineering analysis can now be achieved through the use of machine calculations. The approach to be taken in any particular application depends upon the objectives, the availability of computers, and the qualifications of personnel. In this study a rational approach is made to the use of machine computed data that can be tabulated in advance for the determination of heat fluxes through structural sections. The form of the data

FIG. 1a. . . .SINGLE ROOM ENCLOSURE—PLAN VIEW

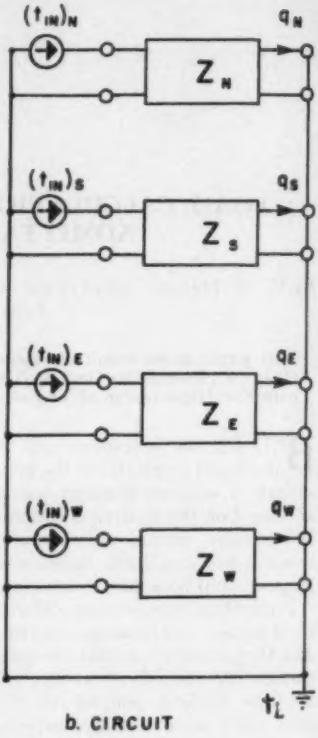

FIG. 1b. . . .SINGLE ROOM ENCLOSURE—CIRCUIT

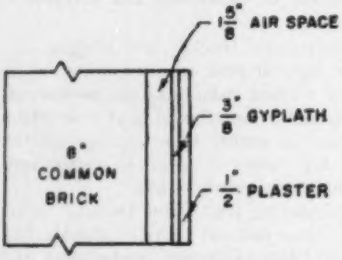

FIG. 1c. . . .CROSS SECTION OF WALL CONSTRUCTION IN EXAMPLE

tabulated is such as to make it independent of the boundary conditions and directly applicable to any situation. Limitations of the calculation procedure to be described are discussed in another section.

Beginning with the thermal network representation of an enclosed space, each conduction path network may be represented by a transfer impedance function t_{in}/q as shown in Reference 6. The transfer impedance thus defined is a function of input frequency and boundary resistance as well as thermal properties of the structure itself. In Reference 6 it was suggested that the transfer function be expressed in terms of the characteristic thermal impedance offered by a given construction and the separate boundary resistances with the view of pretabulating

characteristic impedances for typical constructions. It now appears more desirable to precompute the transfer admittance function q/t_{in} of typical constructions for various values of boundary resistance. This eliminates the necessity of solving the complex algebraic expression for t_{in}/q when making load calculations. Further, it has been demonstrated[7] that transfer functions can be computed and tabulated so economically and rapidly with an a-c electric network analyzer that it would be a relatively small task to precompute and tabulate the transfer admittance

NOMENCLATURE

A = heat transfer area, square feet.

h_i = inside boundary conductance, Btu per (hour) (square foot) (Fahrenheit degree).

h_o = outside boundary conductance, Btu per (hour) (square foot) (Fahrenheit degree).

M = mean-to-peak magnitude of heat flux per unit area at inside boundary per unit temperature difference, $[_{mp}t_{sa} - t_i]$, Btu per (hour)(square foot) (Fahrenheit degree).

q = heat flux at inside boundary, Btu per hour.

R = resistance to heat transfer, Fahrenheit degree per (Btu per hour) (square foot).

R_t = total steady state heat transfer resistance, Fahrenheit degree per (Btu per hour) (square foot).

t_i = inside design temperature, Fahrenheit degrees.

t_{in} = time-variable outside design potential or sol-air temperature, Fahrenheit degrees.

$_mt_{sa}$ = mean value of sol-air temperature, Fahrenheit degrees.

$_{mp}t_{sa}$ = mean-to-peak value of sol-air temperature (1st harmonic), Fahrenheit degrees.

Z = transfer function, t_{in}/q, complex quantity represented by M and β.

Greek Letters

β = phase angle or time lag between resulting heat flux, q, and applied temperature (t_{in}), hours.

θ = hours past midnight.

θ_{sa} = time of occurrence of peak value of sol-air temperature, hours past midnight.

Subscripts

E = east wall.

N = north wall.

S = south wall.

W = west wall.

max = maximum value.

min = minimum value.

functions over a wide range of boundary resistances for all types of construction for which thermal properties are known. The methods used in precomputing transfer functions with an a-c network analyzer and an investigation of the influence of lumping, the importance of sol-air temperature harmonics, the importance of accurately determining the convective conductances, and the effects of absorbed radiation on inside surface temperatures are presented in a companion paper listed as Reference 7.

TABLE 1. . . .TABULATED TRANSFER FUNCTIONS OF WALL SECTIONS FOR USE IN CALCULATION OF COOLING LOADS BY STEADY STATE SINUSOIDAL METHOD

WALL SECTION, WITH $h_0 = 4.0$, $h_1 = 1.65$	R_t	M		β	
8-in. Common Brick Insulated	8.28	0.048		9.91	
		1ST HARMONIC	2ND HARMONIC	1ST HARMONIC	2ND HARMONIC
8-in. Common Brick	3.73	0.121	0.0216	6.8	4.8
12-in. Common Brick	3.39	0.0865		9.11	
4-in. Brick 6-in. Tile	3.37	0.154		6.86	
Stucco-Wood Frame	2.07	0.460		0.94	
10-in. Concrete	5.01	0.0747		7.52	

A METHOD FOR LOAD PREDICTION

Many factors are involved in sensible load calculation including: infiltration; ventilation; heat gain through walls, ceiling, floor and glass areas; inside radiation exchange and internal heat sources. The method of calculation presented here actually considers heat gain through walls only, but it can be extended to include other heat flow paths by the method of super-position. It should be recognized that the method presented is a simplified steady-state sinusoidal solution utilizing a maximum amount of pretabulated information.

TABLE 2. . . .SOL-AIR TEMPERATURE DATA

(Obtained by Fourier analysis following procedure given in Reference 8)

ORIENTATION	m'_{sa}	m'_{sa}		θ_{sa}	
		1ST HARMONIC	2ND HARMONIC	1ST HARMONIC	2ND HARMONIC
South	89	18.0	6.84	13.8	0.73
West	93	27.8	13.6	15.8	7.5

The suggested method for load prediction will be explained through the use of an example problem which considers a single enclosure, shown in Fig. 1*a*. The thermal circuit to be solved is shown in Fig. 1*b*. A cross-section of the wall structure is shown as Fig. 1*c* in the example calculation. The problem is solved by determining the load contribution of each wall separately, based on a constant and uniformly distributed inside air temperature.

To calculate the heat flow through individual wall sections, the designer will need: (1) a knowledge of the wall construction and appropriate transfer functions; (2) the inside design temperature; and (3) the input potential expressed as sinusoidal components of a sol-air temperature applicable to the various orientations of the structure under design. Knowledge of these 3 quantities will enable the designer easily and simply to construct a graph showing the variation in cooling

load with time for a 24-hr period. Values of steady-state resistance and transfer functions computed in Reference 7 for several types of construction are given in Table 1. The transfer admittance functions are represented by the mean-to-peak

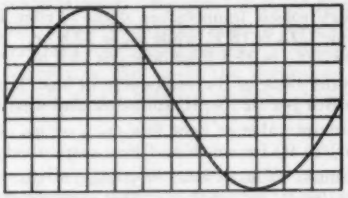

FIG. 2. . . .BLANK GRAPH FOR USE IN PLOTTING RESULTS OF CALCULATIONS

magnitude, M, of heat flux per unit area at the inside boundary resulting from a mean-to-peak magnitude of 1 deg temperature variation applied at the outside boundary. The phase angle, β, in hours is the time lag between the resulting heat flux and applied temperature input. Values of 4.0 Btu per (hr)(sq ft)(F deg) for the outside boundary conductance (h_o) and 1.65 Btu per (hr)(sq ft)(F deg) for

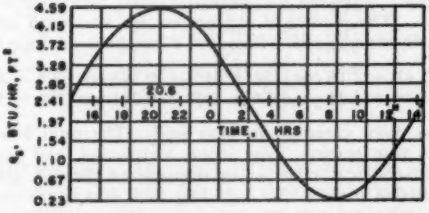

FIG. 3. . . .GRAPH WITH SAMPLE VALUES PLOTTED. DAILY HEAT FLOW THROUGH SOUTH FACING 8-IN. COMMON BRICK WALL

inside boundary conductance (h_i) were used in the computation of steady state resistances and transfer admittance functions.

Sol-air temperatures given in Table 8, p. 302 of THE GUIDE 1957 were used for the sample calculation in order to compare results with calculations based on equivalent temperature differentials. The data given in Table 2 were obtained by a Fourier analysis of the sol-air temperatures following the procedure suggested[8].

The procedure for determining the heat flux for a single wall is as follows with the south and west facing walls being used in this example.

(*a*) For a given wall construction such as 8-in. common brick (Fig. 1*c*), obtain values for R_t, M, and β from Table 1: R_t = 3.73 F deg (hr) (sq ft) per Btu; *1st Harmonic*

Frequency, M = 0.121 Btu per (hr)(sq ft)(F deg); β = 6.8 hr. At 2nd harmonic frequency, M = 0.0216 Btu per (hr) (sq ft)(F deg); β = 4.8 hr.

(*b*) Determine the wall area, A, for each orientation. For this example calculations are made on the basis of 1 sq ft wall area.

(*c*) Decide on the inside design temperature; Let t_i = 80 F.

(*d*) Obtain the necessary sol-air temperature data. From Table 2, ${}_m t_{sa}$ = 89 for south and for 1st harmonic ${}_{mp}t_{sa}$ is 18 F, while θ_{sa} is 13.8 hr. Similarly, ${}_m t_{sa}$ = 93 for west and for 1st harmonic ${}_{mp}t_{sa}$ is 27.8 F while θ_{sa} is 15.8 hr.

(*e*) Calculate the constant value or d-c component of heat flux, $q_{dc} = [{}_m t_{sa} - t_i]/R_t$, and obtain 2.41 and 3.48 Btu per (hr)(sq ft) for south and west respectively.

(*f*) Calculate the maximum value of heat flux, $q_{max} = q_{dc} + M({}_{mp}t_{sa})$, and obtain 4.59 and 6.84 Btu per (hr)(sq ft) for south and west respectively.

(*g*) Calculate the minimum value of heat flux, $q_{min} = q_{dc} - M({}_{mp}t_{sa})$, and obtain 0.23 and 0.12 Btu per (hr)(sq ft) for south and west respectively.

(*h*) Determine the time at which the peak value of heat flux occurs. This is found by adding θ_{sa} and β giving 20.6 hr past midnight for the south wall and 22.6 hr past midnight for the west wall.

(*i*) Compute the daily variation in heat flux for each wall. The contribution of each wall to the total load may now be determined by using Fig. 2 as a preprinted curve and labeling the axes in accordance with the computation of q_{dc}, q_{max}, q_{min} and the values of β and θ_{sa}. It should be recognized that a sinusoidal excitation or input always produces a sinusoidal response resulting in a very simple determination of the response when the input and transfer function is known. Thus, for the south wall example, the values for q_{dc}, q_{max}, and q_{min} are entered on the ordinate of Fig. 2. This is shown in Fig. 3 with the intermediate values added. The time scale is established by placing 20.6 hr at the peak heat flux and marking off the entire scale in hours on both sides of the peak. Sinusoids for additional input harmonics may be determined in the same manner. The analytical expressions for the south and west wall heat fluxes can also be easily determined. Based on 2 input harmonics, they are

$$q_s = 2.41 + 2.18 \cos [15\theta + 51°] + 0.148 \cos [30\theta - 83°]$$
$$q_w = 3.48 + 3.24 \cos [15\theta + 18°] + 0.294 \cos [30\theta - 185°]$$

(*j*) Determine the daily variation in heat load. The total load usually includes contributions due to transmission, solar input, ventilation, and various heat sources such as lights, occupants, and solar power transmitted through glass areas. Curves giving the daily variation in heat flux including solar effects are constructed for each of the walls as shown in the example just given. The ventilation load curve is determined in the usual manner (infiltration effects may be added to this curve). Constant and sinusoidally varying radiation sources may be treated as shown in Reference 2, section on *Radiation Sources*. The final step in the computation is to add the values of q, hour by hour, to obtain the final load curve.

DISCUSSION

Certain limitations in the steady state sinusoidal method of load calculation appear at present. As outlined, the method assumes a constant indoor design temperature. Thus, the designer cannot account for the capacitive effects of interior partitions, furnishings, and materials stored in the space. Likewise, the effects of control settings and various types of thermostats cannot easily be studied. In cases where transients are important, the steady state sinusoidal approach has definite limitations. Under these circumstances the transient response method[9] of network solution or d-c network analyzer solutions[10] are the most promising.

In the specification of cooling systems, the complexity of additional harmonics is not necessary in many instances. Design conditions, particularly sol-air temperature inputs, are arrived at through a combination of experience and weighted weather data. In building design, wall heat fluxes do not usually represent a large portion of the sensible load. Because of these facts it is believed that a sinusoid can be obtained which would represent a specified design condition and result in reasonably accurate load calculations. However, the load contributed by additional input harmonics can easily be computed when necessary. For example, instead of a single graph for each wall, as shown in Fig. 3, it would be

TABLE 3. . . .COMPARISON BETWEEN WALL HEAT FLUXES BASED ON EQUIVALENT TEMPERATURE DIFFERENTIALS GIVEN IN THE GUIDE AND NEW METHOD

	HEAT FLUX, BTU PER (HR)(SQ FT)					
	SOUTH WALL			WEST WALL		
	1958 GUIDE	NEW METHOD		1958 GUIDE	NEW METHOD	
		ONE HARMONIC	TWO HARMONICS		ONE HARMONIC	TWO HARMONICS
8 a.m.	1.08	0.25	0.11	2.16	0.90	1.07
10 a.m.	1.08	0.37	0.25	1.62	0.17	0.05
12 a.m.	1.08	1.04	1.06	1.62	0.32	0.03
2 p.m.	1.08	2.07	2.21	2.16	1.32	1.15
4 p.m.	2.70	3.19	3.31	2.70	2.90	3.02
6 p.m.	4.31	4.11	4.09	3.78	4.63	4.92
8 p.m.	4.31	4.57	4.43	5.40	6.06	6.23
10 p.m.	3.24	4.45	4.33	6.48	6.79	6.67
12 p.m.	2.70	3.78	3.80	6.48	6.54	6.25

necessary to prepare an additional graph for each additional harmonic based on a knowledge of the transfer functions for each additional frequency.

Radiation exchange between interior surfaces can be included in load calculations through a re-iterative procedure as shown in Reference 11. A less cumbersome procedure is the insertion of a resistance network representing radiation exchange and the solution by direct electric analogue techniques given in Reference 10. However, these methods do not lend themselves easily to the simplified hand procedure described in this paper. For the case of constant and sinusoidally varying radiation sources, tables such as Tables 6 and 7 in Reference 7 make it convenient to include the resulting load and to determine the temperature rise of the receiving surfaces.

By inspecting Table 3 one may compare the values of heat flux computed by the suggested method with values based on equivalent temperature differentials given in THE GUIDE 1958 for the same construction and sol-air inputs. A *U*-value of 0.27 was used for the brick wall section consistent with resistances given on pp. 177 and 186 of this edition of THE GUIDE. It will be noted that the peak heat flux predicted by the suggested method using 2 input harmonics is less than 3 percent greater than the equivalent temperature differential method but values at other times differ much more. It would be rather difficult to determine pre-

cisely the reasons for these differences in view of the fact that certain approximations and arbitrary adjustments were made in computing the equivalent temperature differentials[5]. From a practical point of view, the difference in peak values predicted by the 2 methods is insignificant.

The main attributes of the suggested method of load calculation presented in this paper are the following:

Simplicity: Requires only a small amount of calculation utilizing previously machine computed local input data and transfer functions of typical constructions. A simple graphical procedure using preprinted sinusoid curves on a grid may be used to display hourly values and the final load curve determined by summation of hourly data.

Flexibility: Can accommodate local variations in weather through the use of the appropriate sol-air input and surface conductances.

Accuracy: Obtained within the limitations discussed previously. The number of input harmonics required in a given instance depends upon how much of the total load is contributed by the wall heat fluxes, the shape of the sol-air input curve, and the type of construction. Any number of harmonics can easily be handled with this method once the transfer functions have been computed. In many instances a single harmonic is sufficient for equipment sizing purposes. A formula for the number of harmonics to be used in any application is not possible.

Additional refinement in load analysis requires an improvement in the knowledge of the thermal properties of materials and composite wall constructions, of microclimate effects on boundary inputs and resistances, and in the incorporation of more of the influencing factors into the thermal network.

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1683

No. 1683

TRANSFER ADMITTANCE FUNCTIONS FOR TYPICAL COMPOSITE WALL SECTIONS†

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS in cooperation with the Department of Engineering of the University of California, Los Angeles.

SOME EFFORT has been made to develop methods of load calculation that will take into account the time variable climate, capacitive effects of the structure, and internal heat sources or sinks. In the development of a rational calculation procedure, thermal network representation has been shown[1, 2] to be particularly convenient for describing the heat transfer problem to be solved. Various methods of solution of the thermal network have been proposed. These have been summarized in Reference 3. Further, it was suggested[3] that for load calculations it would be most desirable to develop a simple hand computational procedure which utilized the maximum amount of pretabulated, machine computed information. Such a procedure utilizing pretabulated transfer admittance functions is described and its limitations discussed in reference[4], the companion paper to this study. The transfer functions used in those calculations are obtained in advance with an a-c network computer by using sinusoidal wave forms to represent the input or sol-air temperature. Thermal network representation and solution necessarily introduces certain errors and compromises. This investigation presents an evaluation of some of these errors and the variables which affect them.

DESCRIPTION OF INVESTIGATION

As an aid to calculations which depend on thermal network representation and solution, the effects of 4 factors on several typical wall sections are presented.

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† This paper is based on one part of an investigation carried out in partial satisfaction of the requirements for the degree of Master of Science in Engineering.

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[1] Exponent numerals refer to References.

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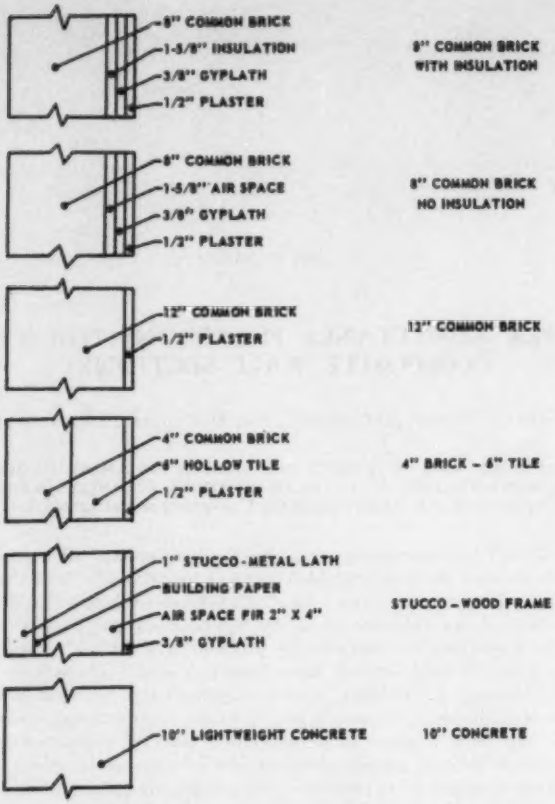


FIG. 1. . . .CONFIGURATIONS AND MATERIALS FOR WALL SECTIONS INVESTIGATED

These factors are, in the order of presentation: (1) errors due to lumping continuous media; (2) the importance of the harmonics of typical diurnal sol-air temperature cycles; (3) the effect of various values of inside and outside surface conductance; and (4) the effect of inside radiation sources.

Five composite wall sections were chosen from the Federal Housing Administration Technical Circular No. 7, January 1947. One additional section, light weight concrete, was used to represent modern, large size structures. These sections are shown in Fig. 1. The network parameters were determined in accordance with the procedure outlined in Appendix B of Reference 2. Table 1 represents the values of resistance and capacitance of the various materials and Table 2 presents the values of surface conductance used in the different network solutions.

The analogous electric networks representing the wall sections were setup on the a-c network analyzer[5, 6] at the University of California. Using this highly flexible and accurate computer, it was possible to investigate the effects of the variables studied on the transfer admittance function by measuring the lag angle and the mean to peak amplitude of output current per unit input potential. Direct measurements of magnitudes, phase angles and vector components of voltages and cur-

TABLE 1. . . .VALUES OF THERMAL AND ELECTRICAL RESISTANCE AND CAPACITANCE USED FOR WALL SECTIONS

WALL SECTION	MATERIAL	THERMAL		ELECTRICAL	
		R	C	R_e	C_e
8-in. Common Brick	Brick	1.59	13.33	2,760	0.667
Insulation	Insulation	5.42	0.271	9,420	0.0136
	Gypsum lath	0.262	0.431	455	0.0216
	Plaster	0.149	0.862	258	0.0432
8-in. Common Brick	Brick	1.59	13.33	2,760	0.667
	Air	0.80	—	1,390	—
	Gypsum lath	0.262	0.431	455	0.0216
	Plaster	0.149	0.862	258	0.0432
4-in. Common Brick	Brick	0.795	6.67	1,380	0.333
6-in. Hollow Tile	Tile	1.57	5.625	2,730	0.2812
	Plaster	0.149	0.862	258	0.0432
12-in. Common Brick	Brick	2.38	20.0	4,125	1.00
	Plaster	0.149	0.862	258	0.0432
Stucco-wood Frame	Stucco	0.0833	2.85	145	0.142
	Paper	0.18	—	313	—
	Air	0.70	—	1,215	—
	Wood	36.0	0.62	62,500	0.031
	Gypsum lath	0.262	0.431	455	0.0216
10-in. Concrete	Concrete	4.15	12.66	7,210	0.633

TABLE 2. . . .VALUES OF THERMAL AND ELECTRICAL RESISTANCES CORRESPONDING TO SURFACE CONDUCTANCES USED FOR WALL SECTIONS

$1/R_i(h)$	R	R_e
0.50	2.000	3472
0.75	1.333	2318
1.00	1.000	1736
1.25	0.800	1387
1.50	0.677	1158
1.75	0.572	892
2.00	0.500	868
3.00	0.333	578
4.00	0.250	434
5.00	0.200	347
6.00	0.167	290
7.00	0.143	248
8.00	0.125	217
9.00	0.111	193
10.00	0.100	174

rent, as well as real and reactive power in any branch of the network, are made by means of 3 major instruments. These 3 instruments, a voltmeter, ammeter, and wattmeter-varmeter, are of the light-beam type, with critically damped moving systems which give exceptionally short times of response. The necessary instrument power is supplied by precision current and voltage amplifiers, which give the metering system a low burden, so low that its insertion into the network under study has a negligible effect on the distribution of currents and voltages in the branches of the network. The overall accuracy of the instrument system, including the voltage divider, the current shunts, and the amplifiers, is within 0.5 percent of full scale.

THERMAL NETWORK SOLUTIONS

The a-c network computer utilizes the thermal-electric analogy[1, 2] and duplicates the thermal network with an identical real electric circuit containing electric elements so chosen that all dimensionless moduli (Pi functions) are equal in magnitude

TABLE 3. . . .SUMMARY OF RATIOS AND UNITS OF ANALOGOUS ELECTRICAL AND THERMAL QUANTITIES

	UNITS		SCALE FACTOR	
QUANTITY	THERMAL	ELECTRICAL	RATIO	VALUE
Time	Hour	Second	θ_e/θ_t	$8.68\ 10^{-5}$
Capacity	Btu/F	Farad	C/C_e	$20\ 10^6$
Resistance	F/Btu/hr	Ohm	R_e/R	1736
Potential	F	Volt	E/t	2
Rate of Energy Transfer	Btu/hr	Ampere	q/i	868

to the dimensionless moduli of the thermal system. Solutions are then obtained by applying electrical potentials and currents corresponding exactly to the thermal circuit boundary conditions, except for scale factors, and measuring the desired quantities. These quantities are then converted to the analogous thermal quantities.

The particular values for the scale factors were restricted by the values of resistance and susceptance◘ available on the analyzer. These are summarized in Table 3.

It is commonplace now in air-conditioning calculations to replace the 3 boundary inputs; absorbed solar radiation, outside surface convection, and longwave radiation exchange with the surroundings, with a single input which results in a time variable temperature, known as the sol-air temperature. Typical sol-air temperatures were used as inputs to the thermal networks. Sinusoids of appropriate amplitude, frequency and phase angle were obtained by means of Fourier analysis and used to represent the sol-air tmperatures. Generators in the network analyzers were used to supply these sinusoids.

Solutions of the network were obtained by applying the appropriate boundary conditions to the network and measuring the resultant phase angles and mean-to-

◘ Susceptance is equal to $2\pi fc$.

peak values of potential and current at points of interest in the network. In this investigation the results are presented using M and β to describe the transfer admittance function, q/t_{in}, or the heat flux per unit input potential. M is defined

NOMENCLATURE

a	= thermal diffusivity, $k/\rho c_p$	sq ft per hr
A	= heat transfer area	sq ft
c	= thermal capacitance	Btu per (F deg)
c_e	= electrical capacitance	farad
c_p	= specific heat capacity	Btu per (lb) (F deg)
E	= electrical potential	volt
f	= frequency of temperature wave	cycle per hr
h_i	= inside convective conductance	Btu per (hr) (sq ft) (F deg)
h_o	= outside convective conductance	Btu per (hr) (sq ft) (F deg)
i	= electrical current	ampere
k	= thermal conductivity	Btu per (hr) (sq ft) (F deg) (ft)
M	= mean-to-peak magnitude of heat flux at inside surface per unit area per unit temperature difference, $(_{mp}I_{sa} - t_i)$	Btu per (hr) (sq ft) (F deg)
q	= thermal current or heat flux	Btu per hr
$(q_s)_r$	= heat flux from inside surface due to net radiation absorbed	Btu per hr
$(q_s)_{net}$	= net radiation absorbed	Btu per hr
R	= thermal resistance	F deg per (Btu per hr) (sq ft)
R_e	= electrical resistance	Ohm
R_i	= inside convective resistance	F deg per (Btu per hr) (sq ft)
R_o	= outside convective resistance	F deg per (Btu per hr) (sq ft)
R_t	= total steady state resistance to heat transfer	F deg per (Btu per hr) (sq ft)
t	= thermal potential or temperature	F deg
t_i	= inside design temperature	F deg
t_{sa}	= sol-air temperature	F deg
$_{mp}t_{sa}$	= mean-to-peak value of sol-air temperature	F deg
t_w	= wall surface temperature	F deg
Δt_w	= change in wall surface temperature	F deg
T_{av}	= average wall surface temperature	R deg
R	= radiosity	Btu per (hr) (sq ft)

Subscripts

$_j$	= j^{th} surface
$_n$	= n^{th} surface

Greek Letters

β	= phase angle or time lag between resulting heat flux (q_s) and applied temperature	hr
ξ	= emissivity	dimensionless
θ_e	= electrical time	second
θ_t	= thermal time	hr
λ	= wave-length	ft
ρ	= density	lb per cu ft
ρ	= reflectivity	dimensionless

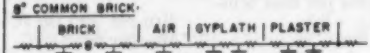
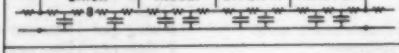
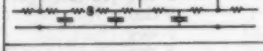
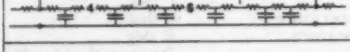
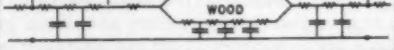
as the mean-to-peak magnitude of thermal current or heat flux per unit area at the inside boundary resulting from a mean-to-peak magnitude of 1 F deg temperature variation applied at the outside boundary. The phase angle, β (hr), is the phase relationship or time lag between the resulting heat flux at the inside boundary and the applied input temperature.

The results of this investigation will be presented under four headings: Lumping, Harmonics, Surface Conductances, and Radiation.

LUMPING

This is the representation of a distributed property system by a finite number of discrete resistances and capacitances. The error introduced by lumping the properties of the system is a function of the fineness of the lumping, the pattern of the

TABLE 4. . . . REFERENCE NETWORKS AND VALUES OF M AND β FOR 3 FREQUENCIES BASED ON THE FIRST THREE INPUT HARMONICS [h_o = 2.0 AND h_1 = 0.5 BTU PER (HR) (SQ FT) (F DEG)]

REFERENCE NETWORKS	FREQ.	M	β
8" COMMON BRICK 	1/24	0.062	8.1
	1/12	0.011	5.7
	1/8	0.004	1.0
8" COMMON BRICK (WITH INSULATION) 	1/24	0.027	11.0
	1/12	0.006	2.1
	1/8	0.002	2.7
12" COMMON BRICK 	1/24	0.035	10.1
	1/12	0.007	1.5
	1/8	0.003	3.9
4" BRICK – 6" TILE 	1/24	0.070	8.0
	1/12	0.014	5.8
	1/8	0.004	0.9
STUCCO-WOOD FRAME 	1/24	0.235	1.8
	1/12	0.096	1.5
	1/8	0.055	1.8
10" LIGHTWEIGHT CONCRETE 	1/24	0.036	9.7
	1/12	0.005	1.4
	1/8	0.002	2.0

lumping, the boundary conditions, the length of time after a change in input occurs, and the location in the network where the value of temperature or heat flux is required. To actually determine the error introduced due to lumping would require the comparison of the network solution to an analytical solution of the distributed system. This is particularly difficult and uneconomic for composite constructions. Instead a standard or reference solution was obtained for each construction by means of a reasonable network representation made finer and finer until no significant change could be detected in the transfer function. The reference networks for the various constructions and the resulting transfer functions for each (in terms of

M and β) are given in Table 4. In order to conserve space the number of capacitors for some sections are indicated numerically rather than drawing them all in the table.

Transfer functions were determined for successively courser networks and for networks representing different lumping patterns for each of the 6 constructions. A value of 2.0 was used for h_o and a value of 0.5 was used for h_i in all of these determinations. Because transfer functions vary in value with input frequency it was of

TABLE 5. . . .SIMPLIFIED NETWORKS AND DEVIATION OF M AND β FROM REFERENCE VALUES

NETWORK CONFIGURATIONS	DEVIATION FROM REFERENCE	
	M, %	β, %
8" COMMON BRICK (WITH INSULATION) a. BRICK | 4 | INSUL. | GYPLATH | PLASTER | b.	-11 -8	-3 -56
8" COMMON BRICK a. BRICK | 6 | AIR | GYPLATH | PLASTER | b.	-9 -11	0 -42
12" COMMON BRICK a. BRICK | 4 | PLASTER | b.	-11 -20	-10 -48
4" BRICK – 6" TILE a. BRICK | TILE | PLASTER | b.	-5 6	-5 -42
STUCCO-WOOD FRAME a. STUCCO | PAPER | AIR / WOOD | GYPLATH | b.	3 -9	-13 65
10" LIGHTWEIGHT CONCRETE a. CONCRETE | 4 | b.	-7 -18	-4 -49

interest to determine the values of M and β for different input frequencies. These are given for the reference networks in Table 4.

Values of M and β for the 8-in. common brick section only were determined for frequencies based on the first 3 input harmonics for successively courser networks. The results indicated that the transfer function is increasingly sensitive to fineness of lumping for higher and higher input frequencies, however, the higher frequencies are less important in determining the heat flux, dropping from a value of M of 0.0617 for the first harmonic to 0.004 for the third harmonic. It was concluded that lumping errors could be assessed by examining the response of the networks to the fundamental frequency only.

It is not feasible to generalize the deviation in the value of the transfer function from the reference value in terms of the number of lumps. For composite structures

TABLE 6. . . . SOL-AIR INPUTS GIVEN TO THREE HARMONICS (FROM REFERENCE 3, θ_t MEASURED FROM MIDNIGHT)

West Wall	$t_{in} - t_i = 0.64 + 32.05 \cos(15\theta_t + 138°) + 18.6 \cos(30\theta_t - 112°) + 9.61 \cos(45\theta_t - 17°)$
East Wall	$t_{in} - t_i = 4.90 + 24.55 \cos(15\theta_t - 142°) + 17.2 \cos(30\theta_t + 135) + 11.6 \cos(45\theta_t - 14°)$
North Wall	$t_{in} - t_i = 8.84 + 13.04 \cos(15\theta_t + 141°) + 3.97 \cos(30\theta_t - 112°) + 1.94 \cos(45\theta_t - 85°)$
South Wall	$t_{in} - t_i = 8.59 + 12.38 \cos(15\theta_t + 142°) + 3.57 \cos(30\theta_t - 101°) + 1.42 \cos(45\theta_t - 94°)$

it is not only the number of lumps that is involved but also the pattern of lumping. In order to save publication space the results for only a limited number of network representations are shown in Table 5. The results of 2 network configurations for each of the different constructions are shown. In each case the (a) network resulted in a deviation of M of approximately 10 percent or less from the reference values. The (b) networks consider the wall sections to have a uniform effective resistance equal to the total steady state resistance and represent the thermal capacitance by a single capacitor.

The results displayed in Tables 4 and 5 can assist greatly in making a judgment regarding the manner in which a network should be drawn to represent a particular construction in a given application. It is of interest to note that in all cases representation of a wall by a single lumped r-c network resulted in quite a large deviation of phase angle (β) from the reference value, whereas the mean-to-peak amplitude (M) was affected in lesser varying amounts for the different constructions.

In Appendix C of Reference 1 it was suggested that the length of conduction path to be lumped could be taken as a certain fraction of the wave-length (λ) of the temperature wave passing through the structure where λ was given as $2\sqrt{\pi a/f}$. Such a criterion is useful for homogeneous walls but cannot be applied to composite construction. For example, the fractions of λ (on which brick lumping was based) for the 3 different constructions using common brick varied from about $1/25\ \lambda$ to $1/12\ \lambda$ for about a 5 percent change in the mean-to-peak amplitude (M) from the reference value. In other words the manner in which a given material is located

and what other materials are present influences the size of lump to be used in meeting a specified accuracy. The best that can be done now is to display data in the manner shown in Tables 4 and 5 to assist in making good engineering judgments. Many factors must be weighed before deciding on a particular network representation. Some of these are economy in the use of computer machine elements, the extent that a particular conduction path (or heat flux) influences the final result

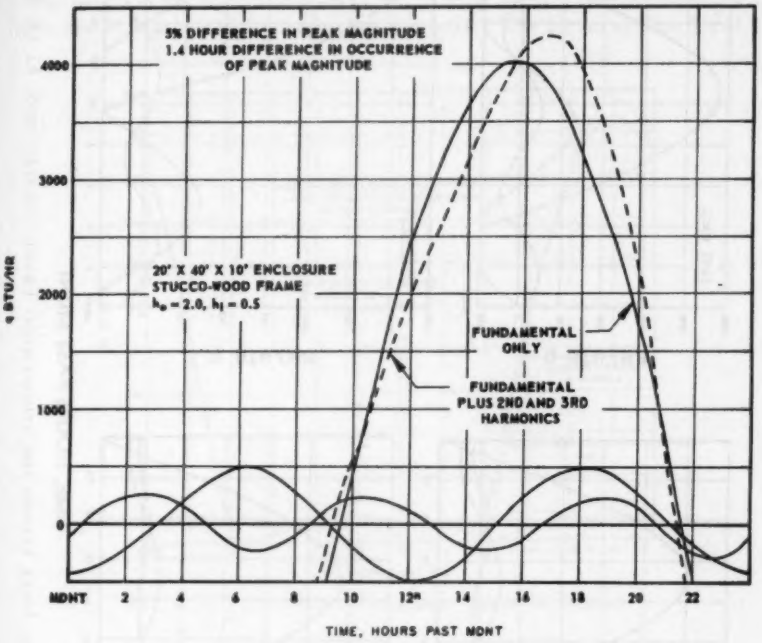


FIG. 2. . . .TOTAL HEAT TRANSFERRED THROUGH WALLS

sought, the importance of phase angle in the study being made and the overall accuracy required. For the determination of transfer functions, a fine network that represents the construction accurately is desirable. This does not present any problem when using a network analyzer of the type employed in this study because sufficient r-c elements are generally available, the increase in labor in setting up the problem is small, and there is no increase in computing time.

HARMONICS

To investigate the effect of input harmonics on the prediction of the total load, a simple structure 40 ft long, 20 ft wide, and 10 ft high was subjected to sol-air temperature inputs given in Table 6. Note that the amplitude of the second harmonic

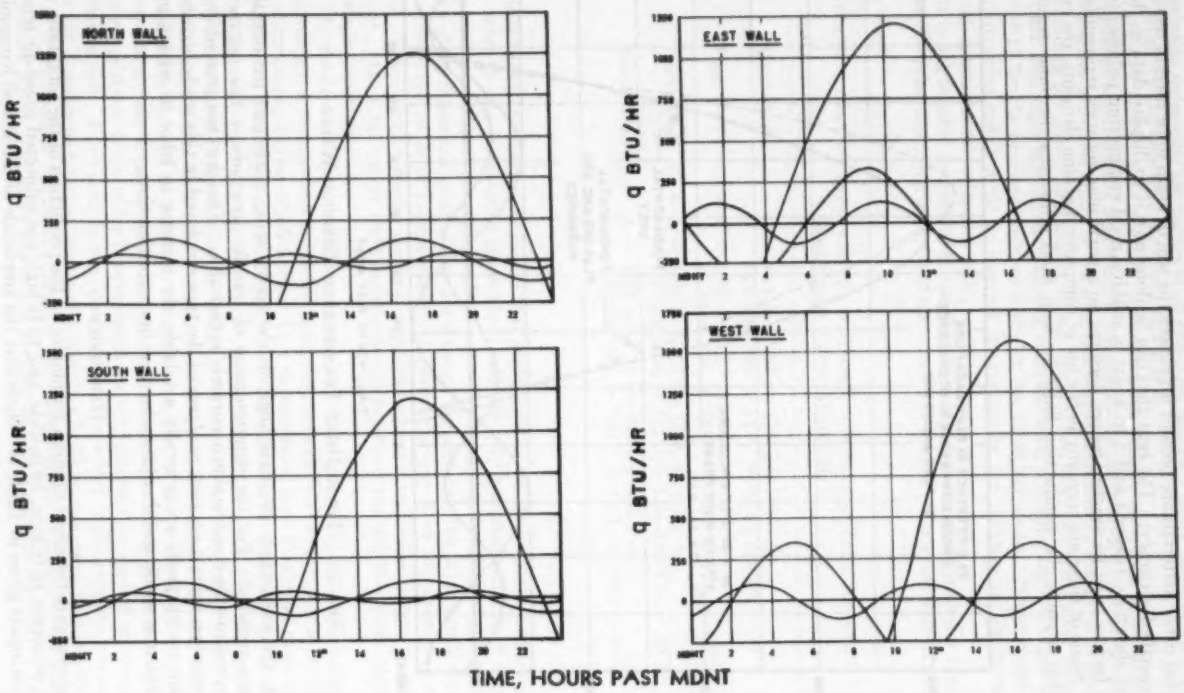


FIG. 3. . . .SINGLE WALL HEAT FLUXES FOR STUCCO-WOOD FRAME CONSTRUCTION, h_o 2.0, h_i 0.5

for the West and East orientations are about 60 and 70 percent of the fundamental amplitude. The structure was oriented so that the short walls faced East and West and it was assumed that heat transfer through the roof was negligible. This is an extreme case where East and West weather walls exert a strong influence on the total load. Only the stucco-wood frame and the uninsulated 8-in. common brick constructions were investigated. These constructions attenuate the higher input harmonics the least. The values for M and β given in Table 4 were used for these calculations.

The results of the calculations are shown in Figs. 2 and 3 for the stucco-wood frame and in Figs. 4 and 5 for 8-in. common brick. The curves in Figs. 2 and 4

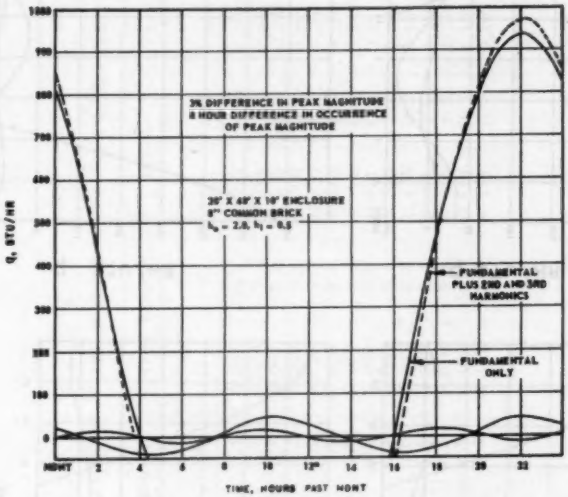

3% DIFFERENCE IN PEAK MAGNITUDE
8 HOUR DIFFERENCE IN OCCURRENCE OF PEAK MAGNITUDE
30' X 40' X 10' ENCLOSURE
8" COMMON BRICK
FUNDAMENTAL PLUS 2ND AND 3RD HARMONICS
FUNDAMENTAL ONLY
MIDNT
TIME, HOURS PAST MIDNT

FIG. 4. . . .TOTAL HEAT TRANSFER THROUGH WALLS

indicate that the peak load calculated for the simple structure based on the fundamental only did not differ very much from the value based on 3 harmonics, 5 percent less for the stucco-wood frame and only 3 percent less for the 8-in. brick construction. The error in the prediction of the time of occurrence of the peak load was substantial for the stucco-wood frame (1.4 hr), but insignificant for the brick construction. As would be expected, the peak heat fluxes for the East and West walls, based on the fundamental only, differed the most from that predicted using 3 input harmonics. A difference of about 20 percent is apparent from Fig. 5.

SURFACE CONDUCTANCES

When the thermal response of a wall section is defined by a transfer function it is of interest to know the influence of inside and outside surface conductances on the value of the function. To study this effect, M and β were measured using the

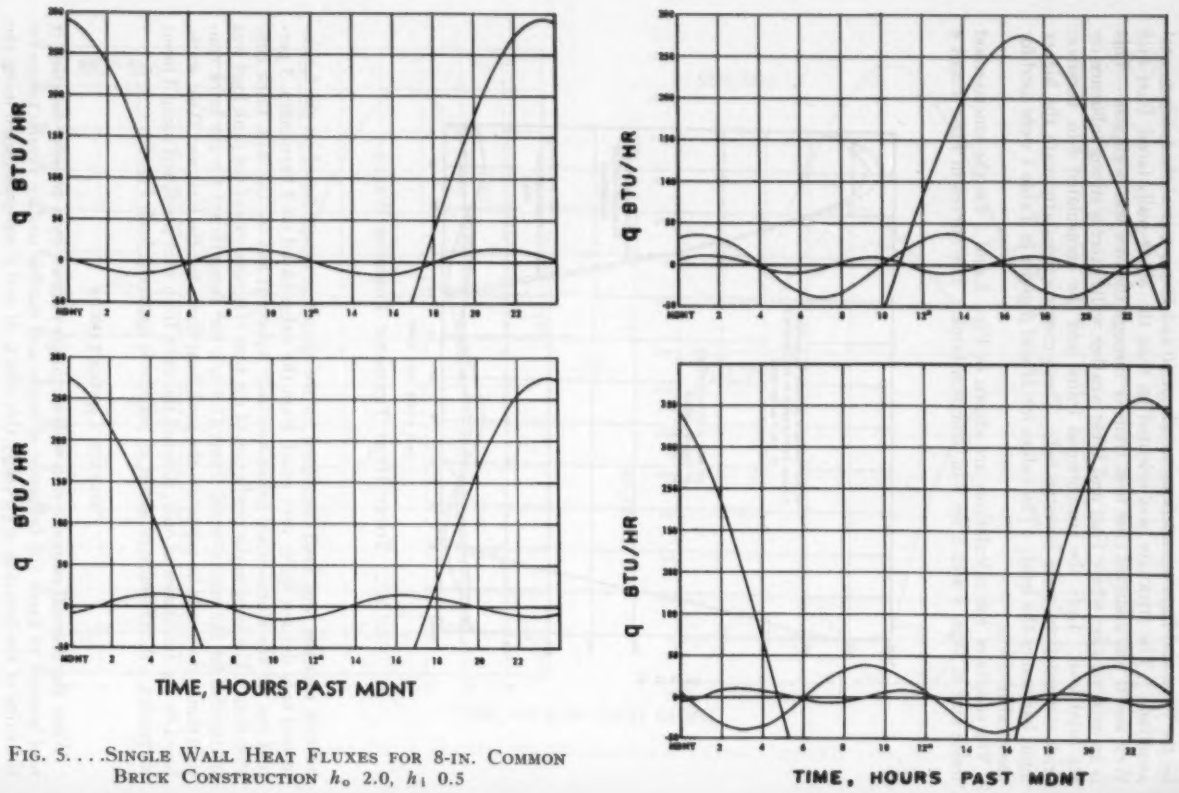


FIG. 5. . . .SINGLE WALL HEAT FLUXES FOR 8-IN. COMMON BRICK CONSTRUCTION h_o 2.0, h_i 0.5

TABLE 7. . . .VALUES OF M AND β FOR VARIOUS FILM COEFFICIENTS

FILM COEFFICIENTS		8-IN. BRICK INSULATION		8-IN. BRICK NO INSULATION		12-IN. BRICK NO INSULATION		STUCCO WOOD FRAME		4-IN. BRICK 6-IN. TILE		10-IN. CONCRETE	
h_0	h_1	M	β	M	β	M	β	M	β	M	β	M	β
2	0.5	0.027	11.0	0.056	8.1	0.035	10.1	0.242	1.6	0.070	8.0	0.0356	9.7
4	0.5	0.034	10.6	0.071	7.6	0.043	9.8	0.274	1.1	0.087	7.5	0.0410	9.4
6	0.5	0.037	10.4	0.078	7.4	0.047	9.6	0.286	0.9	0.095	7.2	0.0436	9.3
8	0.5	0.038	10.3	0.081	7.3	0.049	9.5	0.290	0.8	0.098	7.1	0.0452	9.1
10	0.5	0.040	10.2	0.084	7.2	0.050	9.4	0.296	0.7	0.101	7.0	0.0455	9.1
2	1.0	0.034	10.6	0.080	7.6	0.055	9.8	0.340	1.5	0.106	7.5	0.0532	9.2
6	1.0	0.047	10.1	0.111	6.9	0.076	9.2	0.406	0.8	0.143	6.7	0.0651	8.7
10	1.0	0.051	9.7	0.119	6.7	0.080	9.0	0.421	0.6	0.151	6.5	0.0679	8.6
2	1.5	0.037	10.4	0.094	7.4	0.069	9.5	0.390	1.4	0.126	7.1	0.0630	9.0
6	1.5	0.052	9.7	0.129	6.7	0.095	8.8	0.473	0.7	0.169	6.4	0.0773	8.5
10	1.5	0.056	9.5	0.138	6.5	0.098	8.7	0.491	0.6	0.179	6.2	0.0806	8.4
2	2.0	0.039	10.3	0.104	7.2	0.079	9.4	0.421	1.3	0.139	6.9	0.0689	8.8
6	2.0	0.054	9.7	0.139	6.5	0.108	8.7	0.516	0.7	0.190	6.1	0.0847	8.2
10	2.0	0.059	9.4	0.150	6.3	0.115	8.5	0.535	0.5	0.203	5.9	0.0883	8.2

reference networks for each of the constructions. The outside conductance (h_o) was varied from 2 to 10 Btu per (hr) (sq ft) (F deg) and the inside conductance was varied from 0.5 to 2.0 Btu per (hr) (sq ft) (F deg). The resulting values of M and β based on the fundamental input are displayed in Table 7.

The results clearly indicate a substantial effect on the value of M due to variations in h_o and h_i. As one would expect the magnitude of the effect varies with the type of construction. Except for quite light construction (stucco-wood frame) the effect on the phase angle (β) is small even for large changes in either h_o or h_i. Insulation or a large increase in wall resistance makes M much less sensitive to changes in both h_o and h_i. The sensitivity of M to changes in h_i also decreases for the higher values of h_i.

Radiation

The problem of describing the influence of net radiation gains or loss on the transfer function representing a wall section arises when these functions are used to predict the thermal behavior of a whole system such as a room. Several types of information may be desired.

1. The contribution to the room load of all of the net radiation gains by each of the surfaces.

2. The temperature change of each or certain interior surfaces as a result of a net radiation gain or loss.

3. In the case of radiant heating or cooling panels, the value of the thermal sink or source required to maintain specified space conditions.

It is convenient to consider the net radiation gain of a surface as a thermal current input, $(q_r)_{net}$, as shown in Fig. 6.

For steady radiation inputs the contribution to room load, $(q_2)_r$, may be computed from

$$(q_2)_r = (q_r)_{net} \frac{R_t}{R_t + R_i} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and superimposed on the weather contribution. Also, the rise in temperature of the inside surface (Δt_w) due to absorbed radiation may then be computed from

$$\Delta t_w = R_i(q_2)_r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Values of surface temperature rise and resulting heat transferred to the space are given in Table 8 for various values of net radiation absorbed. The values were computed for the different constructions and are based on $h_o = 2.0$ and $h_i = 0.5$ Btu per (hr) (sq ft) (F deg).

If the net radiation absorbed is time variable, R_t in Equation 1 must be replaced by the characteristic thermal impedance looking into the network at terminal 2, shown in Fig. 6. This characteristic impedance can be determined in a manner similar to the transfer admittance functions and tabulated for different constructions. Time variable sources may be represented by sinusoids and introduced as thermal currents at terminal 2 (Fig. 6). The contributions to the room load may then be computed from Equation 1 by superposition having replaced R_t by the impedance looking in at terminal 2.

In cases when the various surfaces facing each other are at substantially different temperatures (as in the case of panel heating or cooling systems) the net radiation absorbed by any surface $(q_r)_{net}$, is not only a function of radiation sources and sinks but is also a function of the location and arrangement of the surfaces involved,

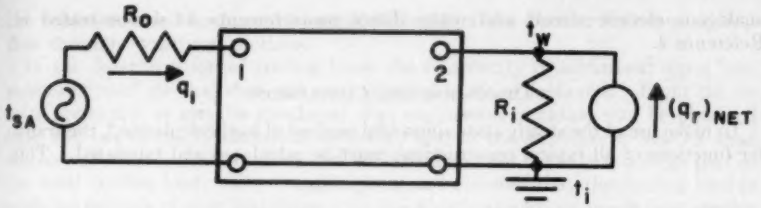


FIG. 6. . . .THERMAL NETWORK REPRESENTING A SINGLE WALL SECTION

radiation properties of the surfaces, and indirectly a function of the construction of the wall itself to the extent that it affects the wall surface temperature. In Appendix C of Reference 2 it was shown that radiation exchange including all interreflections could be represented by a resistance network derived on the basis that the net radiation absorbed could be expressed either as the difference in radiation absorbed and emitted, or as the difference in incident radiation and radiant flux leaving the surface. The thermal current generator shown in Fig. 6 for a single wall section may now be replaced by the resistance network shown in Fig. 7 for the j^{th} surface.

The network for each surface comprises a resistance (R_r) to be connected at terminal 2 which is mainly a function of the radiation properties of the surface and a set of resistances radiating out from this single resistance representing mainly the geometric properties of the enclosure. It should be noted that the values given for $(R_r)_j$ and R_{jn} reflect the approximation that

$$(T_w)_j^4 - (T_w)_n^4 \cong 4[(t_w)_j - (t_w)_n]T^3_{av}$$

This approximation is very good for surface temperature differences encountered in the heating or cooling of spaces.

The radiation exchange network acts as a coupling of all the single wall networks. For this situation the best method to obtain design information is to set up the

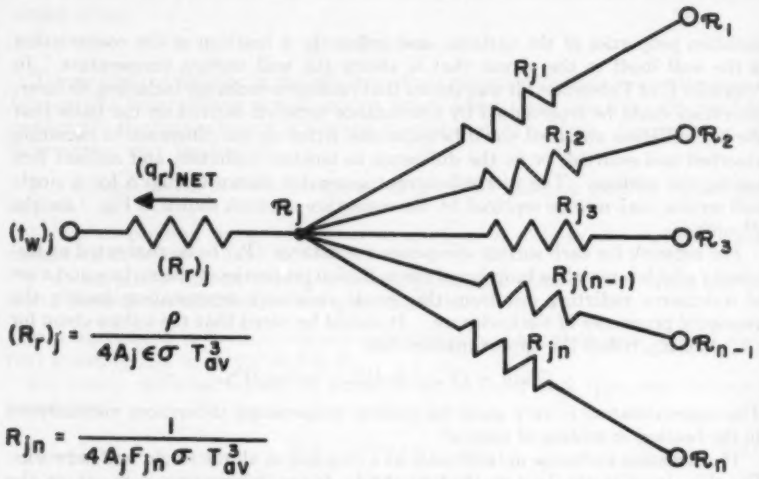
TABLE 8. . . .HEAT FLUX AND WALL SURFACE TEMPERATURE RISE CONTRIBUTED BY STEADY RADIATION SOURCES

TYPE OF CONSTRUCTION		ABSORBED RADIANT ENERGY $(q_r)_{net}$, BTU/(HR) (SQ FT)			
		1.0	2.0	3.0	4.0
8-in. Common Brick	$(q_2)_r$	0.622	1.25	1.87	2.49
No Insulation	Δt_w	1.24	2.49	3.73	4.98
8-in. Common Brick	$(q_2)_r$	0.797	1.59	2.39	3.19
With Insulation	Δt_w	1.59	3.18	4.78	6.37
12-in. Common Brick	$(q_2)_r$	0.601	1.20	1.80	2.40
No Insulation	Δt_w	1.20	2.40	3.60	4.80
4-in. Common Brick	$(q_2)_r$	0.600	1.20	1.80	2.40
6-in. Hollow Tile	Δt_w	1.20	2.40	3.60	4.80
Stucco-Wood Frame	$(q_2)_r$	0.472	0.943	1.414	1.89
	Δt_w	0.94	1.88	2.88	3.76
10-in. Concrete	$(q_2)_r$	0.700	1.40	2.00	2.80
	Δt_w	1.40	2.80	4.20	5.59

analogous electric circuit and make direct measurements as demonstrated in Reference 2.

DISCUSSION AND CONCLUSIONS

To make use of the steady state sinusoidal method of load calculation [4], the transfer functions of all typical constructions must be calculated and tabulated. This



Due to an error in definition in Ref. 2, the values of R_j and R_{jn} given there contain one too many emissivities (ξ) in the denominators. This has been corrected here.

FIG. 7. . . . RADIATION EXCHANGE NETWORK FOR j^{th} SURFACE BASED ON N RADIATING SURFACES[2]

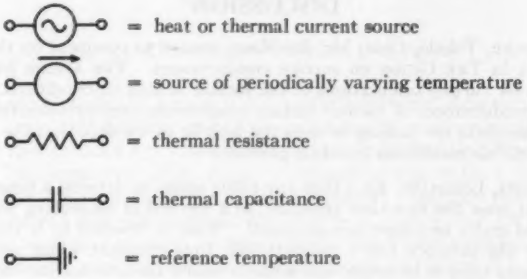
can best be done with a computer such as the one used in this study. It has been found that transfer admittance functions for various constructions can be quickly and easily obtained using an a-c network computer, providing data is available from which reliable values of thermal resistance and capacitance can be calculated. The results shown in the section on lumping indicate that the use of a fine network to represent a structural section is desirable for the determinations of transfer functions. This does not offer any difficulty when using a network computer. After familiarity is gained with the calculation method and computer techniques, one man-hour may be considered sufficient time to complete the transfer functions of one wall section. This includes calculation of thermal and electrical quantities, computer solution and tabulation of results.

Results of the section concerned with the effect of varying convective conductances on the unit mean-to-peak heat flux (M) and lag angle (β) clearly indicate that

the values assumed for h_o and h_i appreciably affect the transfer functions or heat flow through structural sections.

In the determination of cooling loads the complexity of additional input harmonics beyond the first does not seem justified in many instances. From the results presented, it may be concluded that engineering accuracy can be obtained from the use of the fundamental frequency only if the structure is heavier than stucco-wood frame or if the East and West walls do not contribute a large part of the total cooling load. The example presented assumes that the cooling load is made up entirely of wall heat fluxes. In most load calculations inside heat sources such as lighting, human occupants, and infiltration or ventilating air, contribute a substantial part of the load. The smaller the contribution of wall heat fluxes to

SYMBOLS


= heat or thermal current source
= source of periodically varying temperature
= thermal resistance
= thermal capacitance
= reference temperature

the total load, the smaller will be the effect on the load prediction of a given error in estimating that contribution.

For cases where the value of net radiation absorbed $(q_r)_{net}$ can be computed independently for a given surface its contribution to the room load can be determined as shown in the section on radiation. In cases where radiation exchange paths between surfaces are important, it is recommended that they be represented by a resistance network as discussed previously. It is possible that further simplifications of the radiation exchange network could be made that would make computer calculations unnecessary. However, such approximations can only be determined by an examination of many typical examples. This remains to be done in the future.

This investigation has shown the capabilities of an a-c network analyzer in handling thermal network problems and in the study of several variables affecting their solution. The steady state sinusoidal method of load calculation is adapted as a means of utilizing previous electric analogue experience and to provide the designer with a simple calculation tool. This method of calculation is presented in detail in Reference 4, a companion study to this investigation. Utilization of the a-c network computer would be an economical method for computing the transfer functions of all typical constructions in use throughout the country.

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[2] Harry Buchberg: ASHAE RESEARCH REPORT No. 1543—Electric analogue prediction of the thermal behavior of an inhabitable enclosure (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 339).

[3] Harry Buchberg: ASHAE RESEARCH REPORT No. 1647—Cooling load from pretabulated impedances (ASHAE TRANSACTIONS, Vol. 64, 1958, p. 503).

[4] W. B. Drake, Harry Buchberg, and D. Lebell: ASHRAE RESEARCH REPORT No. 1682—Load calculations using pretabulated admittance functions (ASHRAE TRANSACTIONS, Vol. 65, 1959, p. 515).

[5] *The A-C Network Analyzer of the University of California* (University of California Engineering Department, 1950).

[6] *A-C Network Analyzer* (General Electric Co., GEI 28848, May 1949).

DISCUSSION

G. R. MUNGER, Toledo, Ohio: Mr. Buchberg seemed to comment on there being insufficient data in THE GUIDE on surface conductances. THE GUIDE has inside film conductances for various orientations of the surface at still air conditions. It has outside surface conductances of various surface roughnesses and air velocities. Does the author feel that data are lacking or does the lack lie in the fact that the inside film is truly not at still air conditions in actual practice?

R. V. PRUCHA, Louisville, Ky.: Was any effort made to develop a time constant for these walls, at least the first time constant, as a method of estimating how many harmonics should really be taken into account? What is referred to is the very simple diagram that the network has a characteristic time constant which determines how many harmonics need to be taken into account before the attenuation characteristic of the transfer function starts going down. Was any effort made to correlate the data on that basis?

AUTHORS' CLOSURE (Mr. Buchberg): In connection with the remark regarding the lack of data on conductance in THE GUIDE, reference was to the fact that the equivalent temperatures for a non-steady heat transfer situations are based on a single conductance.

There are footnotes that tell how to make corrections. Also in one of the chapters there is information on how to compute conductances as a function of air speed and other variables. However, the calculation procedure can be made more convenient.

In the case of transfer functions, they are readily determined taking into account the surface resistances and can be tabulated in that fashion. All that would be needed would be to pick out a single entry which is appropriate for the particular local condition.

In connection with the determination of a characteristic network time constant as a means of estimating the number of harmonics required, the authors felt at the time that the most direct and simplest method of evaluation was to compare the value of the transfer function (magnitude and lag angle) for several frequencies and finally to determine the actual effect on the total heat flux for an enclosure. The authors were not aware of an easily determined time constant for composite and nonsymmetrical constructions that could be used in a rational manner to determine the number of input harmonics required for a particular situation.

No. 1684

LARGE-SCALE WALL HEAT-FLOW MEASURING APPARATUS†

By K. R. SOLVASON*, OTTAWA, CANADA

THE Division of Building Research of Canada's National Research Council has recently completed construction of an apparatus to measure steady-state and periodic heat flow through 8-ft square wall sections at its Building Research Centre in Ottawa. The apparatus (Figs. 1 and 2) consists of two 8- by 8- by 4-ft boxes open on one side, between which the test wall is placed. One box is maintained at the desired constant cold side temperature from −35 F to +50 F for steady-state tests or varied according to some predetermined cycle for periodic heat flow tests. The other box is electrically heated to maintain a constant warm side temperature of from 65 F to 75 F. The heat transmission coefficient for the wall specimen is calculated from the measured electrical input and temperatures.

In the design of the apparatus an attempt was made to overcome the limitations of the following methods for determining heat transmission coefficients: (*a*) calculations using predetermined thermal conductivities or conductances of the various components and air films; (*b*) the ASTM guarded hot box method; and (*c*) the use of heat flow meters.

LIMITATIONS OF EXISTING METHODS

Many wall combinations are difficult to assess by means of calculation for the following reasons:

1. The only reliable thermal conductivities available for most materials are those obtained by hot plate tests on dry materials which may be considerably different for actual operating conditions.

2. Surface conductances are not necessarily constant over the whole of a wall surface and the average conductance will depend on the temperature difference between the wall surface and air, and that between the wall surface and its surroundings.

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† This paper is a contribution from the Division of Building Research and is published with the approval of the Director.

* Associate Research Officer, Division of Building Research, National Research Council, Canada.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N. Y., June 1959.

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FIG. 1. . . .VIEW OF APPARATUS

3. Most wall combinations contain heat flow paths of differing conductance, the effect of which is difficult to assess without resorting to a rigorous mathematical treatment or to analogue techniques, because of lateral heat flow in some of the components.

The ASTM guarded hot box[1] method consists essentially of placing an electrically heated metering box over the center portion of the warm side of the test wall. Surrounding the metering box is a larger guard box. The temperatures in the 2 boxes are adjusted so that there is no temperature difference (and hence no heat flow) across the walls of the metering box. The conductance is calculated from the electrical input to the metering box and the temperature difference across the wall.

The main disadvantages of this method are: (*a*) The metering box interferes with the convection over the test wall, so that forced convection must be resorted to and this may give film coefficients different from those occurring in practice. It is difficult to produce equal coefficients for the metering area and the guard area, so that lateral heat transfer may occur from the measuring area to the guard area. (*b*) The metering box placed over the central portion of the test wall measures only the heat flow into that portion but it has been shown by G. O. Handegord and N. B. Hutcheon[2] that this is not necessarily the average heat flow for the whole test

[1] Exponent numerals refer to References.

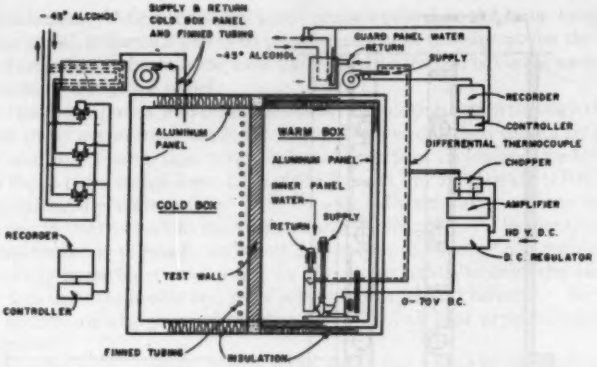

FIG. 2. . . .SCHEMATIC CROSS SECTION OF APPARATUS

wall, particularly for walls containing vertical air spaces; blocking of air spaces in the test area may change the conductance substantially. (*c*) Radiation exchange is indefinite and it is difficult to produce the same effect on both the metering area and the guard area of the test wall. Differences in the radiant exchange from the inner and outer surfaces of the metering box may require that different air temperatures be provided in the test box and the guard box in order to maintain zero heat flow across the test box walls and this may lead to lateral heat transfer in the test wall. (*d*) In many cases the metering box will not cover a representative complete module of the test wall.

The use of heat flow meters of the multiple differential thermocouple type is considered inadequate for the measurement of heat transmission coefficients for the following reasons:

1. The heat flow must be integrated over the whole wall surface and an accurate weighted average is difficult to obtain on walls where large variations in heat flow occur.

2. The meter, if placed on the wall surface, probably interferes with the convective transfer.

3. The thermal resistance of the meter itself reduces the heat flow in the area covered by the meter. Although corrections for this effect can be calculated, for some cases, they are probably not reliable when applied to walls that have surfaces of high conductivity material, since heat transfer in the plane of the wall surface permits part of the heat to bypass the meter.

4. Good contact between the meter and wall surface is often difficult to achieve and air spaces between the meter and wall surface will increase the thermal resistance through the meter-wall combination; heat flows in the plane of the meter to the points of contact are possible so that, if the thermopile element is localized in, say the center of the meter, the heat flow might bypass the element or the heat flow through it might be exaggerated, depending on where contact occurs.

CRITERIA FOR DESIGN OF THE APPARATUS

With these considerations in mind the heat flow apparatus was constructed to meet the following requirements. *First*, it was designed to meter the heat flow into the whole of representative sections of building walls, and *second*, to expose the

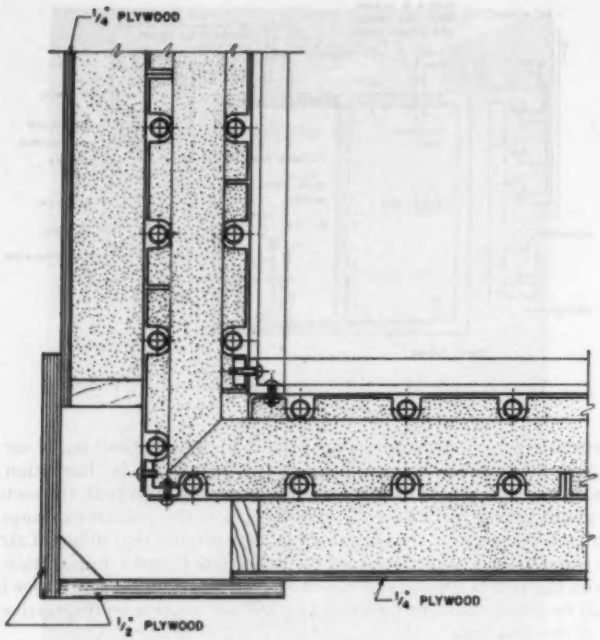

FIG. 3. . . .DETAILS OF WARM BOX WALLS

test wall to controlled temperature surroundings as well as to air at controlled temperature. *Third*, it was built to operate without forced air circulation over the warm side of the test wall in order to produce warm-side film coefficients approaching those occurring in practice. *Fourth*, it operates with the cold-side temperature either constant or varying according to some predetermined cycle.

DESCRIPTION OF APPARATUS

The apparatus consists essentially of a cold box and a guarded hot box combination, with an 8-ft square measuring area. This size was chosen in order to accommodate walls with air spaces 8-ft high and with modules up to 4-ft wide. It was decided to limit the depth of the box to 4 ft for practical reasons. This was considered large enough to produce natural convection effects, approaching those occurring in practice.

The warm box was constructed with a liner of aluminum panels (Fig. 3) similar to those used to line the ASHAE environmental rooms. These panels, which were developed for use in radiant heating and cooling applications, consist of aluminum extrusions with holders for copper tubing, through which liquid is circulated. The inside of the box was painted to increase the emissivity of the panels to about 0.9. A second set of panels was installed outside of the liner and separated from it by

thermal insulation. When the outer panel is maintained at the same temperature as the inner panel, it forms a guard to prevent any heat transfer across the walls of the box. Insulation (2-in. mineral wool batts) and a plywood covering were applied to the outside of the guard panel.

A water reservoir, pump, and pump motor for circulating water through the tubes of the inner panel are located inside the box. The pump is belt driven by a series-wound d-c motor. The voltage-current characteristics of this type of motor permit the energy input to be varied from 25 watt (85 Btu per hr) to 500 watt (1700 Btu per hr) by regulating the voltage from 15 to 90 volt. Direct-current power was considered easier to control and to measure accurately than a-c. The electrical input to the pump motor is normally sufficient to provide all of the heat requirements; thus the energy imparted to the water by the pump serves to heat the panel and the energy loss from the motor and drive serves to heat the air directly. Air heaters and water heaters are also provided so that the ratio of air heat to panel heat may be varied if desired.

Twelve parallel water circuits (each circuit supplying 4 by 4 ft of panel) are used to supply the panels in order to permit the use of a large quantity of water without excessive pump head and pump power. At the maximum input of 1700 Btu per hr, about 20 gpm of water are circulated. Since only about 60 percent or less of the total power input will be supplied to the water, the drop in temperature passing through the panel is only about 0.085 F. The ratio of water quantity to input increases as the total input decreases, so that the temperature drop through the panel is always less than 0.085 F. It can also be shown that the panel to water temperature difference and also the temperature variation over the panel surface is quite small, not more than 0.1 F.

The water reservoir is incorporated into a shield in order to prevent the test wall or the box walls from *seeing* the pump motor, which will operate at some temperature higher than the panel.

A second reservoir and pump located outside of the warm box supplies water to the outer guard panel. The reservoir contains a cooling coil and an electric heater to control the water temperature.

The cold box is simply a well insulated box with an aluminum panel liner and a sheet metal exterior vapor barrier covering. The panel liner serves a three-fold purpose: (*a*) to provide controlled temperature surroundings to which the test wall is exposed; (*b*) to provide some convective heat exchange area; and (*c*) to intercept heat gains to the box itself and thus reduce the heat exchange area required inside the box. Additional heat exchange area (312 sq ft surface area of finned tubing) is provided to bring the air temperature as close as possible to the panel temperature. Liquid (⅓ water, ⅓ ethylene glycol and ⅓ methyl alcohol) at the desired temperature is circulated through the tubes of the panel liner and the finned tubing. This liquid is cooled in a shell and tube heat exchanger by a low temperature liquid (−45 F methyl alcohol) from a central low temperature chiller serving other laboratories in the Centre. The heat exchanger contains eight ⅜-in. outside diam copper coils 20 ft long. The control valves are arranged so that the cold liquid can be circulated through 2, 4, 6, or all 8 coils depending on the temperature difference desired.

CONTROLS AND MEASURING INSTRUMENTS

The control and measuring instruments for the apparatus were selected in order to limit any specific error to as low a value as practicable, or to a maximum of one

percent at the minimum heat flow and temperature difference. A heat flow of 200 Btu per hr and an overall temperature difference of 30 F were selected as a design basis. Errors in the determination of an overall heat transmission coefficient may arise from errors in measurement, in control of temperatures and power input, and from heat leakage.

Measurement of Temperatures: Temperature measurements are made with 30-gauge copper-constantan thermocouples and an electronic self-balancing temperature indicator. Checks of the indicator and thermocouple wire indicate that an accuracy of ± 0.05 F can be obtained. The temperature difference measurements are thus accurate to about ± 0.1 F deg. The percentage error is then about ± 0.33 percent at an overall difference of 30 F.

Measurement and Control of the Heat Input, and Warm Box Temperatures: Warm box control system is required to adjust the input voltage to the warm box to a stable value such that the heat input equals the wall transmission within 2 Btu per hr or one percent, and to maintain the guard panel at a temperature such as to prevent a heat transfer of more than 2 Btu per hr across the box walls. The panel liner of the warm box and the water have a large thermal capacity, about 330 Btu per F deg. The maximum permissible temperature change is, therefore, only 0.006 F deg. per hr in order to limit heat storage to 2 Btu per hr.

The insulation between the inner and guard panels provides a calculated conductance of approximately 0.125 Btu per (hr) (sq ft) (F deg) or a total heat flow of 24 Btu per (hr) (F deg). Thus for precision of one percent, the average temperature difference must be controlled to about ± 0.08 F deg. A moderate ripple in the outer panel temperature can be tolerated, provided the average temperature is within the ± 0.08 F deg, since the insulation between the inner and outer panel will provide sufficient damping to prevent any appreciable heat flow at the inner panel.

The control system is shown in block form in Fig. 1. A thermocouple actuated electronic recorder with a 3-action (proportional + reset + rate) controller is used to control the guard panel temperature. This controller regulates a solenoid cooling valve and electric heater with manual switches, so that heating and cooling can be supplied continuously on, continuously off, or pulsed on-off by the controller. The d-c input to the inner panel pump is supplied by a d-c power supply[3], the output voltage of which is regulated from 0 to 70 volt by the amplified unbalance from a differential thermocouple to maintain equal inner and guard panel temperatures. The differential thermocouple signal is first converted to a square wave by a mechanical chopper, amplified by a high gain low level d-c amplifier and rectified by the chopper. The higher voltage d-c signal is then used to regulate the output voltage of the power supply.

This system has been used with only moderate success. Slight fluctuations in the guard panel temperature produce large fluctuations in the inner panel power. The effect of rapid fluctuations has been eliminated successfully by applying thermal damping to the guard panel thermocouple junction. Lower frequency temperature swings produce a similar swing in inside power. This effect, which is not serious for steady-state tests but unsuitable for periodic tests, could be eliminated by converting the present thermocouple recorder-controller to a shorter span and more accurate resistance-element-type.

Some additional difficulty has, however, been experienced with the chopper-amplifier system. The chopper point adjustment is difficult to maintain over long periods and this, together with stray inductive pickup, produces fluctuations in the power and errors difficult to assess. The present power supply, in addition, is not

of sufficient capacity for the maximum inputs required. The control and power system is therefore being replaced with equipment which has recently become available. A magnetic amplifier of 500-watt maximum capacity is being procured for a power supply. The magnetic amplifier regulated by the present recorder (converted to a resistance type) and a current adjusting controller will be used to control the inner panel temperature. The recorder sensitivity will be approximately 0.001 F deg and the combination is expected to control to 0.002 to 0.003 F deg. The error due to the capacity effect of the box is thus expected to be no more than 0.5 percent at the minimum heat flow.

The guard panel temperature will be controlled by an on-off electronic temperature controller which has a sensitivity of 0.001 F deg and is expected to control temperatures to well within ± 0.03 to 0.04 F deg of the inner panel. The unbalance error is then expected not to exceed about 0.5 percent.

The input voltage and current are measured by a millivolt recorder connected across appropriate resistors. The maximum recorder error, with calibration, can be limited to less than ± 0.25 percent of full scale so that by sizing the resistors to give larger than half-scale deflections this error should always be less than 0.5 percent. The error in voltage times current is less than 1.0 percent.

Control of Cold Box Temperatures: No special problems were encountered in the control of the cold box temperature. Here a thermocouple-actuated electronic recorder with a 3-action temperature controller is used to control 1, 2, or 3 solenoid valves which regulate the flow of cold liquid through the heat exchanger. The number of valves used is selected manually, depending on the temperature, to make ¼, ½, ¾ or all of the heat exchanger area effective. With this combination, the cold-side temperature can be held constant to about ± 0.1 F deg or about 0.33 percent of the temperature difference.

The controller is now equipped so that an approximate sine wave variation in temperature can be produced with amplitudes of 5 to 15 F deg and periods of 6, 12 or 24 hr. A program controller may be added later with which any desired wave form can be produced.

If all the measurement and control errors were additive, the total error is expected to be less than 2.0 percent. Operation to date indicates that heat transmission coefficient values can be reproduced within 1.0 percent. Tests to assess measurement errors experimentally will be conducted in the near future.

EDGE HEAT LEAKAGE

Heat leakage at the edge of the test wall is another source of error. The apparatus is located in a laboratory where the temperature is maintained close to the usual warm-box operating temperature. Heat leakage will, therefore, tend to occur through the edge of the test wall into the cold box. The edges of the test wall will usually be surrounded with low conductivity material ($k = 0.25$) slightly thicker than the test wall and some 5½ in. wide (the width of the box edges) as shown in Fig. 4. Aluminum angles attached to the cold box panel and extending 2 in. over the box edge are installed to extend the low temperature plane outside of the test wall area. A similar angle is attached to the guard box panel so that the heat loss for this portion will be supplied by the guard panel rather than the inner panel.

The additional thickness of insulation (x in Fig. 4) should be such that its thermal resistance approximately equals the air-film resistance, so that the temperature in the plane of the wall surface will equal wall surface temperature. In many cases, the arrangement of the edges of the test walls may be such that good edge guarding

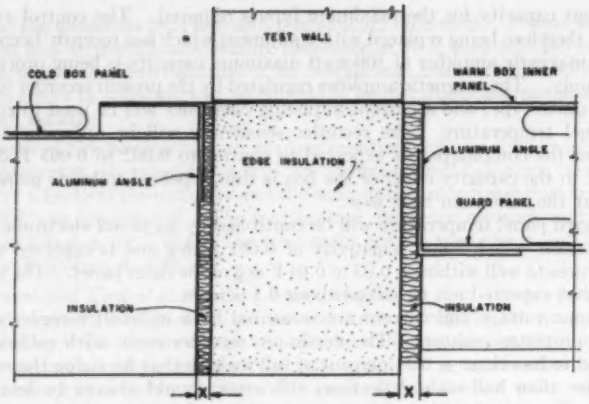

FIG. 4. . . .ARRANGEMENT AT TEST WALL EDGES

is very difficult. It may therefore be necessary to use test sections smaller than 8 ft square and use a filler of known thermal properties.

It would be quite difficult to calculate the error introduced by edge leakage, except perhaps by one of the relaxation or analogue methods. Tests conducted to date indicate no significant temperature gradients on the test wall surfaces near the edges, so that this error can be assumed to be rather small. Further tests to assess edge effects will be carried out in the near future.

OPERATING CHARACTERISTICS

The total heat transfer at the warmside surface of the test wall will be the sum of the net radiant exchange from the box panels to the test wall plus the convective exchange from the air to the test wall.[4] This can be expressed as:

$$q_t = q_r + q_c = F_e F_c \times 0.173 \left[\left(\frac{T_p}{100} \right)^4 - \left(\frac{T_w}{100} \right) \right] + x(t_a - t_w)^y \quad . \quad . \quad (1)$$

The radiant and convective components are usually accounted for by coefficients h_r and h_c respectively, such that

$$q_t = h_r(t_p - t_w) + h_c(t_a - t_w) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The usual surface conductance coefficient, f_1, is the sum of h_r and h_c, hence,

$$q_t = f_1(t_1 - t_w) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

If the average air temperature (t_a) in the apparatus is not equal to the panel temperature, t_p, an equivalent temperature t_1, can be calculated from Equations 1, 2 and 3 which would produce the same heat flow as the actual t_p and t_a. Thus,

$$t_1 = \frac{h_r t_p + h_c t_a}{h_r + h_c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The form factor F_c for the test wall and box can be shown to be equal to 1.0, and the emissivity factor F_e to equal $e_p e_w$ where e_p and e_w are the emissivities of the box walls and the test wall surface respectively.

If e_p and e_w are known, the factors for convective heat flow x and y in Equation 1 can be evaluated by measurements of panel temperature, t_p, average test wall temperature, t_w, and average air temperature, t_a, at several values of heat transmission, q_t. For the range of conditions expected, these factors are not expected to vary so that they need be evaluated for one wall only. In any later tests, if the average wall surface temperature can be measured, it will be possible to calculate the wall surface emissivity. For those walls where surface temperature variations make the assessment of the average difficult, it will be possible to calculate the average temperature if the emissivity e_w is known. The cold-side heat exchange can be treated similarly.

Thus the surface conductance can be accurately established, the average surface temperatures defined, and the conductance of the wall itself (without surface conductances) can be calculated. Overall conductances for any other surface conductances can then be calculated.

NOMENCLATURE

q_t = total heat flow rate, Btu per (sq ft) (hr).
q_r = net radiant exchange, Btu per (sq ft) (hr).
q_c = convective exchange, Btu per (sq ft) (hr).
F_e = the form factor for radiant exchange between the box panels and the test wall.
F_e = emissivity factor.
T_p = box panel temperature, Fahrenheit absolute.
T_w = average test wall temperature, Fahrenheit absolute.
t_a = average air temperature, Fahrenheit.
t_p = box panel temperature, Fahrenheit.
t_w = average wall surface temperature, Fahrenheit.
t_i = temperature, Fahrenheit, equivalent to t_a and t_w.
x = constant.
y = constant.
h_r = surface coefficient for radiant heat transfer Btu per (hr) (sq ft) (F deg).
h_c = surface coefficient for convective heat transfer Btu per (hr) (sq ft) (F deg).
f_i = combined surface coefficient Btu per (hr) (sq ft) (F deg).

The warm side surface conductance, f_i, resulting in the apparatus is expected to be quite close to the still air conductances that result in practice, if a wall is exposed to air and surroundings at nearly equal temperatures. This coefficient is not a constant but rather a function of the temperature level, the temperature difference, and the wall emissivity.

The cold-side surface conductance coefficients in the apparatus are expected to be much lower than will be desired in most cases. Forced circulation either by fans or induced by jets will be provided to increase the conductance as required.

CONCLUSION

This apparatus is expected to provide an accurate and realistic means for obtaining steady-state heat flows or conductances through built-up wall sections, windows, and doors. The sample size used in the apparatus is considered large enough so that the results, particularly for inside surface conductances, will be representative of much present-day construction.

The test wall is exposed to controlled temperature surroundings as well as to controlled temperature air. The convective and radiant components of the surface conductances can thus be evaluated.

The apparatus also provides a means for assessing the transient response of wall sections. In this respect the effect of moisture movement induced by cycling temperature can be investigated.

The apparatus can also be adapted to general calorimetry tests such as the lag characteristics and maximum air-conditioning loads from appliances. Tests for water vapor and air transfer can also be carried out without any major additions or alterations.

ACKNOWLEDGMENTS

The author is indebted to N. B. Hutcheon, assistant director of the Division of Building Research, and to A. G. Wilson, head of the Building Services Section, for their many helpful suggestions in the design of the apparatus and in the preparation of this paper, and to H. L. Egan, laboratory assistant, who performed most of the construction work.

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[1] *ASTM Designation* C236-54T: *Tentative Method of Test for Thermal Conductance and Transmittance of Built-Up Sections By Means of The Guarded Hot Box* (American Society for Testing Materials, 1954 Supplement to Book of ASTM Standards, Part 3, p. 204).

[2] G. O. Handegord and N. B. Hutcheon: Thermal performance of frame walls (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 171).

[3] T. M. Dauphinee and S. B. Woods: Low-level thermocouple amplifier and a temperature regulation system (*The Review of Scientific Instruments*, July 1955, p. 693).

[4] W. H. McAdams: *Heat Transmission* (McGraw-Hill Book Co., Inc., New York, New York, 1942, 2nd ed., p. 459).

DISCUSSION

HARRY BUCHBERG, Los Angeles, Calif.: The author has presented an excellent paper on the design and development of a very well conceived and useful piece of apparatus. As pointed out, many different studies can be made with its use. Of particular interest would be the determination of transfer functions in order to check the validity of calculations that are based on a knowledge of the properties of the materials that go into the structure.

As another comment the impression was gained that the author does not have very much confidence in the use of heat meters. His attention might be drawn to the work of Professors Gier and Dunkle at the University of California in the determination of wall conductances using heat meters, in which they solved at least many of the difficulties mentioned by the author.

J. G. MACORMACK, New York, N. Y.: Does the equipment permit orientation of direction of heat-flow? Is it possible to run the equipment in a horizontal position and get the heat-flow downward?

AUTHOR'S CLOSURE: Gier and Dunkle's paper is familiar and use has been made of quite a number of their meters and this is one of the reasons for the remarks made in the paper.

With minor modification the equipment can be turned over and used for floor tests or it can be turned over the other way and used for roof and ceiling heat-flow. This would require some modification in the circulation system, but it is just a matter of getting the pump sitting in the right location.

1685

No. 1685

CONDENSATION BETWEEN THE PANES OF DOUBLE WINDOWS†

By A. Grant Wilson* and E. S. Nowak**, Ottawa, Ont., Canada

THE VALUE of double windows in reducing heat transmission through window areas and in permitting higher inside relative humidities during the winter, without excessive condensation on inside glass surfaces, is well known. Some form of double-window arrangement is used in most houses in regions having low winter temperatures. There is an increasing use of double windows in commercial and industrial buildings, particularly when year-round air conditioning is employed.

Condensation of water vapor between the panes, and on the inside surface of the outer pane, is a common occurrence with most types of double windows except the factory-sealed variety. A small amount of such condensation is accepted generally as inevitable. When it begins to obstruct seriously the view through the window for long periods, however, or when the run-off contributes to the deterioration of surrounding materials, there is reason for concern.

Condensation will occur on the inside surface of the outer pane whenever the temperature of that surface at any point is below the dew-point temperature of the air-vapor mixture in the space between the panes. This ultimately will occur, with outside temperatures lower than inside, if the gain in water vapor to the space is greater than the loss.

In this paper the factors involved in the transfer of water vapor to and from the space between the panes of double windows other than the factory-sealed type are considered. A relatively simple approach is developed for determining under what conditions condensation will occur in a given window, or alternately, for designing a window to be free of condensation under given conditions.

Water Vapor Transfer by Diffusion

Water vapor generally moves in to and out of the air space of a double window under two forces. It moves through the materials and cracks in the window assem-

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* Head, Division of Building Research, National Research Council of Canada.

** Division of Building Research, National Research Council of Canada.

Presented at the Annual Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Lake Placid, New York, June 1959.

TABLE 1....RELATIVE VAPOR PRESSURE DIFFERENCES ACROSS INNER AND OUTER PANES

OUTSIDE TEMPERATURE, F	RATIO OF VAPOR PRESSURE ACROSS INNER PANE TO THAT ACROSS OUTER PANE
40	7
20	9
0	18
−20	29
−40	51

bly by diffusion as a result of differences in partial pressure of the water vapor, and it is transferred as a component of the air which flows through the cracks in the assembly under total air pressure differences.

Most of the materials used in window construction are relatively impermeable to vapor, except wood and it can be made relatively resistant through painting. The cracks and openings in the assembly will thus usually provide the major paths for vapor flow by diffusion. The permeability coefficient for air water-vapor mixtures is independent of relative humidity but does vary somewhat with temperature. For purposes of this discussion, however, it can be assumed constant. The vapor flow by diffusion in to and out of the air space is then directly proportional to the pressure difference across inner and outer panes.

The vapor pressure differences across inner and outer panes are dependent on inside and outside temperatures and relative humidities. Table 1 illustrates the possible order of these pressure differences. The values have been calculated taking inside room and air-space vapor pressures corresponding to saturation at the temperatures of the inner surfaces of the inner and outer panes respectively. Outside vapor pressures were taken to correspond to saturation at the outside air temperature. Surface temperatures were calculated for an inside air temperature of 70 F assuming an overall U value for the window of 0.53 and inside and outside

TABLE 2....RELATIVE PRESSURE DIFFERENCES ACROSS LEEWARD WINDOWS DUE TO WIND

WIND SPEED, (MPH)	PRESSURE DIFFERENCE (0.3 VELOCITY HEAD), (IN. OF WATER)
0	0
5	0.004
10	0.014
15	0.032
20	0.058
25	0.090
30	0.130

TABLE 3. . . .PRESSURE DIFFERENCES ACROSS WINDOWS DUE TO CHIMNEY ACTION

OUTSIDE TEMPERATURE, F	PRESSURE DIFFERENCE PER FOOT FROM NEUTRAL ZONE, (IN. OF WATER)
40	0.000867
20	0.00151
0	0.00220
−20	0.00295
−40	0.00378

surface conductances of 1.5 and 6.6 respectively (based on heat transmission values for windows in THE GUIDE 1958 of ASHAE).

The vapor pressure difference across the inner pane is several times greater than the vapor pressure difference across the outer pane in all cases; the ratio of these pressure differences increasing with decreasing outside temperature. Thus, to maintain outflow equal to inflow, the effective resistance of the inner glazing to vapor flow by diffusion must be many times that of the outer glazing.

TOTAL PRESSURE DIFFERENCES ACROSS WINDOWS

Several factors may contribute to total pressure differences across windows. Wind is one commonly recognized. Air movement around a structure will result in a pattern of pressures that depends on many factors. Information on the actual pressures around buildings is relatively sparse. It can be assumed in the simplest cases that the pressures on leeward walls are below barometric while those on windward walls are above it. Pressures inside the building resulting from wind action will depend on the distribution of air leakage throughout the building enclosure. If the leakage characteristics of all exposures are similar, the pressures inside will probably be less than barometric. As a basis for discussion, pressures on the windward side, leeward side, and inside of 0.8, minus 0.5 and minus 0.2 velocity heads respectively can be assumed. This leads to total pressure differences of 1.0 velocity head across windward windows and 0.3 velocity head across leeward windows.

TABLE 4. . . .MINIMUM RATIOS OF OUTSIDE TO INSIDE AIR FOR SPECIFIED CONDITIONS

OUTSIDE TEMPERATURE, F	RATIO OF OUTSIDE AIR TO INSIDE AIR, Q_o/Q_i
40	7
20	9
0	16
−20	25
−40	41

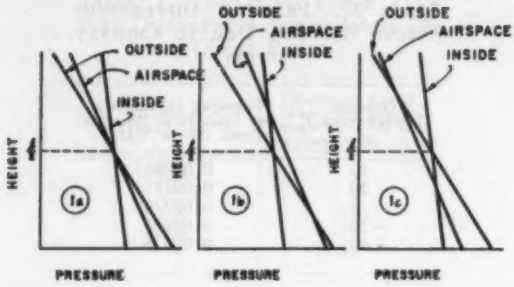

FIG. 1. . . .PRESSURE DISTRIBUTIONS FOR DOUBLE WINDOW AT THE NEUTRAL ZONE

For purposes of heat loss calculations, the flow from outside to inside resulting from the higher pressure difference is significant. It is the flow from inside to outside occurring under the lower total pressure difference, however, that is significant in connection with condensation between double windows, since this results in a net gain in water vapor to the air space. Table 2 illustrates the possible order of the pressure differences across leeward windows as a result of wind action, based on 0.3 velocity head and an air density of 0.075 lb per cu ft.

A second major factor contributing to total pressure differences across windows is chimney action between air in a building and the outside induced by temperature differences. As a result, air tends to flow into the building through lower openings and leave through upper openings. With no other forces acting, there is a neutral zone somewhere between at which inside and outside pressures are equal. The pressure difference across the walls of any enclosed space due to chimney action can be calculated easily providing the temperature differences and the level of the neutral zone are known. Unfortunately, information on neutral zone locations for both residential and commercial buildings is extremely limited.

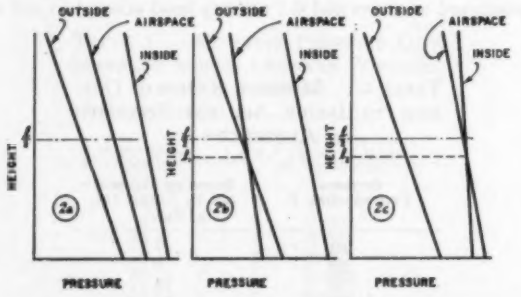

FIG. 2. . . .PRESSURE DISTRIBUTIONS FOR DOUBLE WINDOW ABOVE THE NEUTRAL ZONE

Available experimental records for residences[1] suggest that the actual levels are considerably higher than would be predicted on the basis of the vertical distribution of window and door cracks. In single-story houses, the neutral zone may be above first floor windows, while in two-story houses it may be at the level of second story windows. Recently published information on pressure differences across entrances

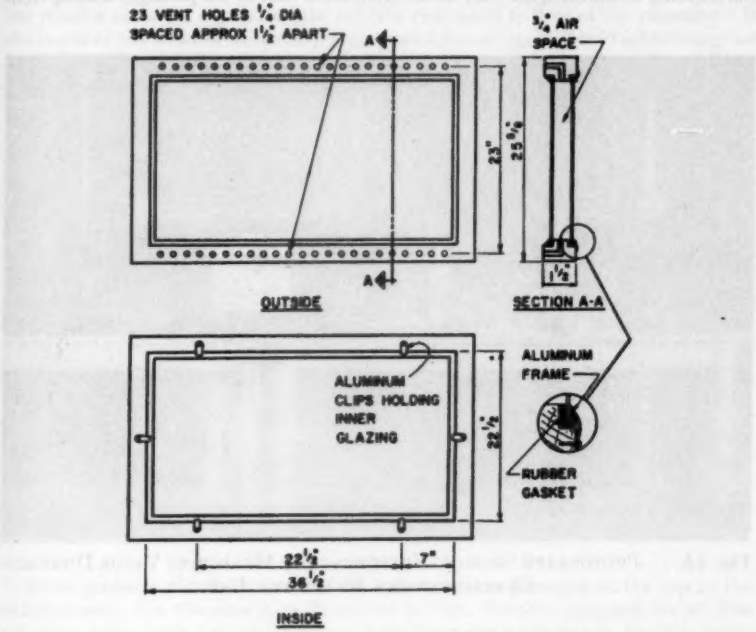

FIG. 3. . . .DETAILS OF CENTER WINDOW

of tall buildings suggests that the neutral zone location in such structures may be well above mid-height.[2]

Since flow from the building to outside occurs only above the neutral zone, the level of the neutral zone is significant in connection with condensation between the panes, windows above the neutral zone being much more likely to exhibit such condensation. Any factor tending to raise the neutral zone, such as venting or mechanical exhaust systems, will reduce the possibility of window condensation; any factor tending to lower the level of the neutral zone, such as pressurizing of buildings, will have the opposite effect.

The order of the inside-outside pressure differences resulting from chimney effect can be seen in Table 3 which gives pressure differences per ft of distance from the

[1] Exponent numerals refer to References.

neutral zone at different outside temperatures. The values are based on inside air at 70 F with a density of 0.075 lb per cu ft.

TOTAL PRESSURE DISTRIBUTIONS ACROSS DOUBLE WINDOWS

In double windows the resistance to air flow of the passages leading from inside the building to the air space may be different from that of the passages leading from

FIG. 4A. . . . PHOTOGRAPH SHOWING EQUIPMENT FOR MEASURING VAPOR DIFFUSION CHARACTERISTICS READY FOR USE

the air space to outside the building. This is not significant with respect to heat loss, where overall leakage values are important. The relative air leakage characteristics of passages in to and out of the air space may be highly significant, however, with respect to condensation between panes. This is apparent when the distribution of total pressures and resulting air flows across double windows are considered.

Fig. 1 illustrates the variations in pressure with height across a double window, subjected to a temperature gradient where the neutral zone occurs at mid-height. In representing the relative slopes of the pressure distribution curves, it was assumed that the temperature of the air space was closer to outside temperature than to that inside. The pressure distributions shown in Fig. 1*a* will result whenever the resistance to flow above and below the neutral zone, around either or both panes, is equal. For example, this will occur with the passages uniformly distributed around the periphery of one or both panes.

Under these circumstances the air-space pressure is between inside and outside pressures at all levels. Thus, inside pressure is higher than air-space pressure above

the neutral zone and lower below the neutral zone. Similarly, the air-space pressure is higher than the outside pressure above the neutral zone and lower below the neutral zone.

The air flow resulting from these pressure distributions will depend on the locations and resistances of the passages. Flow will vary from zero through passages located at the neutral zone to a maximum through passages at the top and bottom of the window. The air space can interchange air with both the inside and outside; the relative amounts depend on the relative resistance to flow of the passages. If the inside of the window were completely sealed the air space would interchange air with the outside only.

Fig. 1*b* illustrates pressure distributions when passages of the outside pane at the bottom have much lower resistance to flow than other passages. Similarly, Fig.

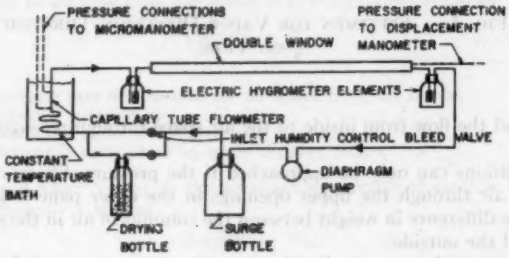

FIG. 4B. . . .DIAGRAM SHOWING ARRANGEMENT OF EQUIPMENT FOR MEASURING VAPOR DIFFUSION CHARACTERISTICS

1*c* shows pressure distributions with the low resistance passages at the top of the outside pane. For the situations illustrated in Figs. 1*b* and 1*c*, appreciable air flow will occur when there are openings past both inner and outer panes, the flow being from inside to out in the former and from outside to in in the latter.

Fig. 2 illustrates the variations in pressure with height across a double window subjected to a temperature gradient where the neutral zone is below the window, i.e., where the total pressure inside is greater than the total pressure outside at all levels.

Fig. 2*a* represents the pressure conditions when the resistances to flow around both panes are equal. The air-space pressure is approximately mid-way between inside and outside pressures and flow is from inside to out through all openings. The air space does not interchange air with the inside or outside. Fig. 2*b* represents the pressure distribution when the resistance to flow through passages of the inner pane is much greater than that through passages of the outer pane, these being evenly distributed above and below the mid-height of the window, with equal openings top and bottom. Under these conditions, the pressure at the top of the air space is greater than outside and the pressure at the bottom is less. The air space can interchange air with the outside, all outflow passing through the upper passages of the outer pane. This consists of both the inflow through the lower openings of the

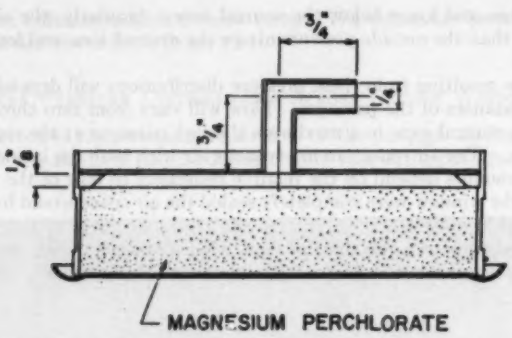

FIG. 5. . . .SPECIMEN FOR VAPOR DIFFUSION THROUGH ¼-IN. TUBE

outer pane and the flow from inside to the air space through passages of the inner pane.

These conditions can only be approached if the pressure difference required to move all the air through the upper openings in the outer pane is less than that created by the difference in weight between the columns of air in the space between the panes and the outside.

Fig. 2*c* represents the pressure distribution when the resistance to flow through the passages of the outer pane is much greater than through those of the inner pane, the passages of both panes being equally distributed above and below the mid-height

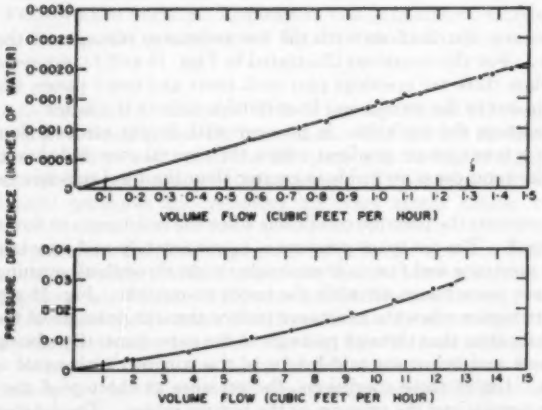

FIG. 6. . . .AIR FLOW *vs.* PRESSURE DIFFERENCE FOR INNER PANE USED IN VAPOR DIFFUSION MEASUREMENTS

of the window. Under this condition, the air space will interchange air with the inside, the amount of air flow through the outer pane being equal to the difference in inflow and outflow past the inner glazing.

WATER VAPOR TRANSMISSION BY AIR FLOW

The foregoing discussion of pressure distributions across double windows has shown that, with the air-flow resistance of the inside pane sufficiently higher than that of the outside pane, and with the openings in the outside pane located at the top and bottom of the window, the air space can interchange air with the outside even when an overall flow from inside to outside occurs. The relationship between these air flows and condensation between the panes can be shown by a simple mass balance.

If vapor flow by diffusion is neglected, the net gain of water vapor by the air space as a result of air flow from the inside is approximately equal to

$$Q_i d_i (W_i - W_s) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

Q_i = volume rate of flow into the air space from the inside.
d_i = density of inside air.
W_i = humidity ratio of inside air.
W_s = humidity ratio of air in space between panes.

The net loss of water vapor by the air space as a result of air flow from the outside is approximately equal to:

$$Q_o d_o (W_s - W_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

Q_o = volume rate of flow into the air space from the outside.
d_o = density of outside air.
W_o = humidity ratio of outside air.
W_s = humidity ratio of air in space between the panes.

To prevent condensation the net gain in water vapor by the air space must equal the net loss of water vapor by the air space.

Equating Equations 1 and 2 it can be shown that,

$$R = \frac{Q_o}{Q_i} = \frac{d_i\,(W_i - W_s)}{d_o\,(W_s - W_o)} = \frac{T_o(W_i - W_s)}{T_i(W_s - W_o)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

R = minimum ratio of outside to inside air to prevent condensation between panes.
T_o = absolute temperature of outside air, Rankine degrees.
T_i = absolute temperature of inside air, Rankine degrees.

The values of the minimum ratio of outside to inside air to prevent condensation at different outside temperatures will depend on inside temperatures and relative humidities. Values given in Table 4 are based on humidity ratios for inside, air space, and outside corresponding to saturation at the temperature of the inner and outer panes and of outside air respectively, using the same conductances as for Table 1. The value of the minimum ratio increases with decreasing outside temperature.

Another factor that produces total pressure differences between the air space and outside is a change in air-space temperature. If the window is not sealed this pres-

sure difference will cause air flow to or from the air space and the window *breathes*. The total amount of air interchange can be calculated from the perfect gas laws and will depend on the magnitude of the temperature change and the volume of the air enclosed in the space between the panes. Whether the air space will lose or gain water vapor as a result of this mechanism will depend on whether the major openings are around the outer or inner panes.

DESCRIPTION OF TEST WINDOW

Laboratory studies were carried out, based on the foregoing considerations, to determine the degree of venting of the air space to outside required to overcome condensation between the panes of a simple wood sash window. Details of the unit are given in Fig. 3. The inner glazing, contained in a metal frame with a rubber gasket, was fixed to the wood sash with aluminum clips while the outer glazing was sealed to the sash with glazing compound. The resistance to air and water-vapor flow around the outer pane was altered by opening or closing ¼-in. diam vent holes provided in the top and bottom members of the sash. The vent holes were approximately 1½ in. long, measured along their center-lines.

WATER VAPOR DIFFUSION AND AIR FLOW MEASUREMENTS

In discussing the mechanisms of vapor transmission in to and out of the air space of double windows, transfer by diffusion and by air flow have been dealt with separately. Both mechanisms will operate simultaneously, however, and will interact in an actual installation, making it impossible to determine their relative contribution to vapor flow. Separate measurements were therefore made of the water vapor diffusion and air flow characteristics of the cracks around the inner pane and of the vent holes around the outer pane as a means of assessing the relative importance of the two mechanisms.

Fig. 4 shows the arrangement used for measuring the vapor diffusion characteristics of the cracks around the inner pane. There were no vent holes around the outer pane in the sash used for this study and the wood was sealed with wax to make it essentially impermeable. The procedure consisted of supplying dry air to the window air space at a known rate and measuring the humidity rise. If there is no total pressure difference between the space and outside, vapor transfer is by diffusion only. At equilibrium, the amount of vapor transferred by diffusion through the cracks around the inner glazing is equal to the gain in moisture of the air flowing through the window space. Thus,

$$M = D\Delta p_v = Qd_a(W_1 - W_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where

M = rate of water vapor transfer past the inner glazing, grains per hour.
D = vapor diffusion coefficient for the inner glazing, grains per (hour) (inches mercury).
Δp_v = difference in partial pressure of water vapor across the inner glazing, inches mercury.
Q = volume rate of flow of dry air being circulated, cubic feet per hour.
d_a = density of air being circulated, pounds per cubic foot.
W_1 = humidity ratio of circulating air at inlet to the window space, grains per pound dry air.
W_2 = humidity ratio of circulating air at the outlet from the window space, grains per pound dry air.

The humidity of the air being circulated was measured at the inlet and outlet of the window space with 2 electric hygrometer elements. These were calibrated prior

to and following the test in an apparatus providing controlled conditions with an absolute accuracy of ±0.1 percent relative humidity. The conditions in the room in which the tests were carried out were controlled at 72 F and 50 percent relative humidity within close tolerances.

The air was circulated by a small diaphragm pump and the rate was measured with a capillary tube flowmeter in conjunction with a sensitive micromanometer. A displacement-type micromanometer in conjunction with a single pan analytical balance was used throughout the test to measure the total pressure difference between the window space and the laboratory. A change in the balance reading of 1 milligram corresponds to about 0.0001 in. of water. Since there was no change, it can be concluded that the total pressure difference across the inner glazing was essentially zero throughout the test.

The vapor diffusion coefficient, based on Equation 4, using the average of initial and final calibrations for the electric humidity sensing elements was 1.89 grains per (hr) (in. Hg). The possible error in the measurement is about ±20 percent.

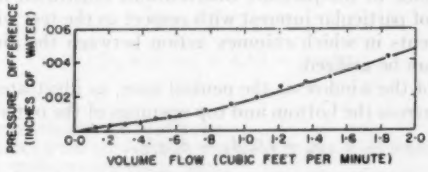

FIG. 7. . . .AIR FLOW *vs.* PRESSURE DIFFERENCE FOR OUTSIDE PANE WITH 46 VENT HOLES

Rather than attempt to make direct measurements of the vapor diffusion coefficients of the ¼-in. diam vent holes, tests were carried out on a ¼-in. diam copper tube with 90 deg bend, mounted in a permeability cup as shown in Fig. 5. The tests were carried out in a conditioned cabinet maintained at 73 F and 50 percent relative humidity. The open end of the tube was protected from direct impingement of air being circulated within the cabinet. The permeance obtained for the tube as shown in Fig. 5 was 90 grains per (hr) (sq ft) (in. Hg). Tests were also carried out on a straight ¼-in. diam tube 1½ in. long and the permeance obtained was 100 grains per (hr) (sq ft) (in. Hg). The corresponding permeability is 150 grains per (hr) (sq ft) (in. Hg per in.).

Air flow tests were carried out on the inner glazing to determine its leakage characteristics during the vapor transmission measurements. The method consisted of supplying air to the window air space at the rate required to maintain a given pressure difference across the window. Pressure differences greater than 0.005 in. of water were measured with a micromanometer sensitive to about 0.001 in. of water; pressure differences less than this were measured with the displacement manometer and analytical balance sensitive to 0.0001 in. H_2O. Subsequent calibration of this manometer has shown that the error in measurement at pressures less than 0.005 is about plus 15 percent. Air flows less than about 2 cu ft per hr were made with the capillary tube flowmeter previously described. At higher flows a calibrated variable area flowmeter was used.

The results of the air flow measurements on the inner glazing are shown in Fig. 6. The density of the air during measurements was essentially 0.075 lb per cu ft. The

air flow was found to be independent of the direction of flow at pressures less than 0.015 in. of water.

Measurements were made to determine the air-flow characteristics of the ¼-in. diam vent holes in the sash around the outer pane, using the equipment just referred to. The flow through the 46 vent holes was not exactly 46 times that through the single vent hole since the holes varied in resistance, having been drilled by hand. Fig. 7 shows the flow characteristics of 46 vent holes in the range of pressure differences normally provided by chimney action between the air space and outside. All flows have been converted with reference to air at 0.075 lb per cu ft.

TOTAL PRESSURE RELATIONSHIPS

To assess the relative importance of vapor flow by diffusion and by air flow in relation to condensation between the panes of double windows, it is necessary to consider what total pressure differences are available across the inner and outer panes at different conditions of overall inside-outside pressure difference. This can be done with reference to the pressure distributions illustrated in Figs. 1 and 2. Figs. 1*a* and 2*b* are of particular interest with respect to the test window since they represent arrangements in which chimney action between the air in the window space and outside can be utilized.

With the center of the window at the neutral zone, as illustrated in Fig. 2*a*, the pressure difference across the bottom and top openings of the outer pane is,

$$p_o = l/2(d_o - d_s)g/g_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where

p_o = pressure difference across top or bottom of outer pane, pounds per square foot.
l = height of window space, feet.
d_o = density of outside air, pounds per cubic foot.
d_s = density of air in window space, pounds per cubic foot.
g = acceleration of gravity, 32.2 ft per (second) (second).
g_c = proportionality constant, 32.2 (mass lb) (ft) per (force lb) (sec) (sec).

This can be expressed as

$$h_o = 3.82 l \frac{T_s - T_o}{T_s \times T_o} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where

h_o = pressure difference across top or bottom of outer pane with window at neutral zone, inches of water.
T_s = mean temperature of air space, Rankine degrees.
T_o = outside air temperature, Rankine degrees.

Similarly the pressure differences across the inner pane at the top or bottom with the window at the neutral zone is,

$$h_i = 0.00721 l \frac{T_i - T_o}{T_s} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where

h_i = pressure difference across top or bottom of inner pane, inches of water.
T_i = inside air temperature, Rankine degrees.

With the center of the window above the neutral zone as illustrated in Fig. 2*b*, the window air space pressure depends on the relative resistance to air flow of cracks

or openings around the inner and outer panes. Flow from outside to the air space occurs through the openings around the outside pane below level l_1. This volume, together with the air flow through cracks around the inner pane, flows through openings around the outer pane above level l_1. If the ratio of outside air to inside air is that defined by Equation 3 then,

$$Q_T = Q_o + \frac{Q_o}{R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where

Q_T = flow through cracks around outer pane above level l_1.

But Q_T and Q_O can be expressed in terms of measured flow characteristics. For the test window the relationship is simple since, as shown in Fig. 7, flow is directly proportional to pressure difference at the small pressure differences produced by chimney action and the vent holes are located only at top and bottom. Thus,

$$Q_o = C_o(d_o - d_s)l_1 g/g \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and

$$Q_T = C_o(d_o - d_s)(l - l_1)g/g_o \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

where

C_o = flow coefficient for vent holes.
l_1 = distance from the bottom vent holes to the level where air space and outside pressures are equal, feet.

Substituting Equations 9 and 10 in Equation 8

$$l_1 = \left(\frac{R}{2R + 1}\right) l \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

It follows, with reference to Fig. 2*b* that

$$h'_o = \left(\frac{2R}{2R + 1}\right) h_o \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

where

h'_o = pressure difference across bottom of outer pane with window above neutral zone, inches of water.

Also

$$h'_i = h - h_o\left(\frac{1}{2R + 1}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

where

h'_i = pressure difference across center of inner pane with window above neutral zone, inches of water.
h = inside-outside air pressure difference across center of window, inches of water.

COMPARISON OF VAPOR TRANSFER BY DIFFUSION AND AIR FLOW

To assess the relative rates of vapor flow by diffusion and air flow in to and out of the air space of the test window, outside conditions of 0 F and 85 percent relative

humidity and inside conditions of 70 F and 25 percent relative humidity have been chosen. The vapor pressure in the space has been taken to correspond to saturation at the temperature of the inside surface of the outer pane. This temperature was calculated assuming an outside surface conductance of 4.0 (for a 10 mph wind) and a U value of 0.51. For these conditions R in Equation 3 equals 3.42.

Based on the measured diffusion coefficient, the vapor transmission by diffusion through the cracks around the inner pane is 0.23 grains per hr. The vapor transmission by diffusion through two ¼-in. vent holes, assuming one at the top and bottom, is 0.0019 grains per hr. Calculated vapor transmission rates by air flow for different conditions of inside-outside total pressure differences are given in Table 5. These values are based on the measured air flow characteristics and the simple relationships developed in the preceding section.

It can be seen that the relative importance of vapor transmission by diffusion and air flow through cracks around the inner glazing depends to a considerable extent on

TABLE 5. . . .CALCULATED VAPOR TRANSFER BY AIR FLOW FOR WINDOW USED IN DIFFUSION TEST

TOTAL PRESSURE DIFFERENCE (IN. OF WATER)	VAPOR TRANSFER THROUGH CRACKS AROUND INNER PANE, (GRAINS PER HR)	VAPOR TRANSFER THROUGH ONE ¼-IN. VENT TOP AND BOTTOM, (GRAINS PER HR)
neutral zone at center $h = 0$	0.515	0.29
neutral zone at bottom edge $h = 0.002$	2.34	0.26
$h = 0.004$	5.1	0.26
$h = 0.014$	15.7	0.26

the inside-outside total pressure difference. With the neutral zone at the center of the window, the vapor transfer by air flow is about twice that by diffusion; with the neutral zone at the bottom edge the ratio is about 10 for the conditions chosen. At higher total pressure differences, vapor transfer by diffusion becomes relatively insignificant. The vapor transfer through the ¼-in. vent holes by diffusion is unimportant compared to that by air flow, regardless of the total pressure difference, so long as the vents are distributed so that chimney action can occur between the air space and outside.

The amount of venting of the air space to outside required to maintain vapor outflow equal to inflow, and thus to avoid condensation between the panes, can be determined from Table 5 for the conditions chosen. With the neutral zone at the center of the window, 2 or 3 vent holes at top and bottom are required depending on the contribution of diffusion to vapor transfer in to the space. With the neutral zone at the bottom edge 10 vent holes at top and bottom are required.

At pressure differences of 0.004 and 0.014 respectively, some twenty and sixty ¼-in. vents are required. These pressure differences were chosen from Table 2 to represent those that might occur across leeward windows with winds of 5 and 10 mph. If vapor transfer from the air space to outside by diffusion is the only available mechanism, the number of vents required is many times the values just given.

Air flow in to and out of the window space as a result of *breathing* action due to temperature changes has been mentioned as a means of vapor transfer. It can be

shown for the test window that the average air flow in to or out of the space due to a daily outside temperature cycle from −20 F to +20 F is about 0.001 cu ft per hr. This compares with a calculated air flow through the cracks around the inner glazing as a result of chimney action, with the window at the neutral zone, of 0.42 cu ft per hr. Thus, for the test window, the amount of vapor transfer as a result of *breathing* action is insignificant relative to that as a result of chimney action. The importance of *breathing* action in moving water vapor in to or out of the air space of a double window will depend on its air leakage characteristics and the volume of air between the panes. Unless the air space is unusually thick, *breathing* action will have only a secondary effect on all but extremely tight windows.

COLD ROOM TESTS

In predicting for the test window the amount of venting of the air space to outside required to overcome condensation under specified conditions, a number of simplifications were introduced. For example, in determining the stack action between the air space and outside it was assumed that the air space temperature could be taken as equal to the mean of the temperatures of the two panes. Both horizontal and vertical temperature gradients in the space were disregarded.

TABLE 6. . . .STUDIES OF WINDOW VENTING IN COLD ROOM

ROOM CONDITIONS	DURATION OF TEST, (HR)	ARRANGEMENT OF ¼-IN. VENT HOLES	OBSERVATIONS
Cold room at 0 F Warm room at 70 F and 38% relative humidity	26	3 top and bottom	borderline
	74	4 top and bottom	borderline
	92	5 top and bottom	adequate
	124	6 top and bottom	adequate
	90	23 bottom	inadequate
	24	23 top	inadequate

Cold room tests were undertaken to determine by direct observation the amount of venting required under specific conditions. For this study, a test window was installed in a wall panel which formed part of a partition between a warm and cold room. The temperature of the cold room could be controlled over a wide range within ±0.25 F deg, with floor to ceiling temperature gradients not exceeding 1 F deg. The relative humidity in the cold room was about 50 percent. Air movement over the window was downward at about 6 mph. Close control of temperature and humidity was also provided in the warm room.

With a warm room temperature of 70 F at the window level, observations of condensation between the panes were made at a number of cold room temperatures and warm room humidities, with various degrees of venting of the air space to the cold room. Results obtained with a cold room temperature of 0 F and a warm room humidity of 38 percent are given in Table 6. None of the conditions covered by Table 6 led to more than light deposits of frost. For the 2 venting conditions judged adequate, the frost was limited to small patches in the two upper corners. The vents were located near the center of the top and bottom edges. It is thought that if they had been more uniformly dispersed the window would have remained

entirely clear. For the two venting arrangements judged to be borderline, there were two or three small additional patches of light frost near the bottom of the window. With the openings at top or bottom only there were two light bands of frost from top to bottom on either side of center.

The window unit used for the cold room tests was a different specimen from that used for the vapor diffusion measurements. Measurements were made to determine the air flow characteristics of the cracks around the inner pane and the results are given in Fig. 8. This unit was found to be somewhat tighter than the other specimen, as can be seen from Fig. 6. It was found that the degree of tightening of the screws holding the aluminum clips affected the tightness of the window. The

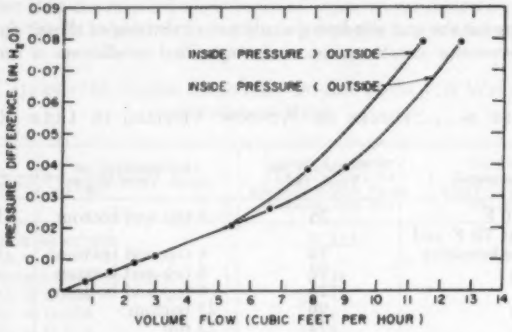

FIG. 8. . . . AIR FLOW *vs.* PRESSURE DIFFERENCE FOR INNER PANE USED IN COLD ROOM STUDIES

air flow characteristics shown in Fig. 8 are believed to be representative of the unit as installed in the cold room.

Measurements of pressure difference between cold and warm rooms were made to determine the total pressure distribution over the test window. Results of these measurements with the cold room at 0 F, given in Fig. 9, indicate that the neutral zone was between the center and bottom edge of the window.

No measurements of the vapor diffusion characteristics of the cracks around the inner glazing used in the cold room tests were made. Predictions of the amount of venting required to prevent condensation between the panes must therefore be based on air flow alone. Such calculations show that with the neutral zone at the center of the window, 1½ vent holes top and bottom are required, while 4 vent holes top and bottom are needed with the neutral zone at the bottom edge of the window. The vapor pressure in the air space was assumed to correspond to saturation at the minimum temperature measured on the inside surface of the outer pane. The comparison of vapor transfer by diffusion and air flow for the inner glazing used in the diffusion measurements has shown that diffusion contributes a significant proportion of the vapor flow at these small pressure differences. Calculations for this unit under the cold room conditions show that the flow by diffusion is from one-third to one-half that by air flow, with the window at the neutral zone. Applying this proportion to the window unit tested in the cold room, the number of vent holes

required becomes 2 and 4½, with the neutral zone at the center and bottom edge respectively.

There is fair agreement between the cold room observations and the estimated amount of venting of the air space required to overcome condensation between the panes. The cold room tests were not ideal for such a comparison because of the quite small total pressure differences provided, when accurate measurements of pressure and flow become difficult. Furthermore, small changes in the location of the neutral zone will lead to relatively large changes in the venting requirements.

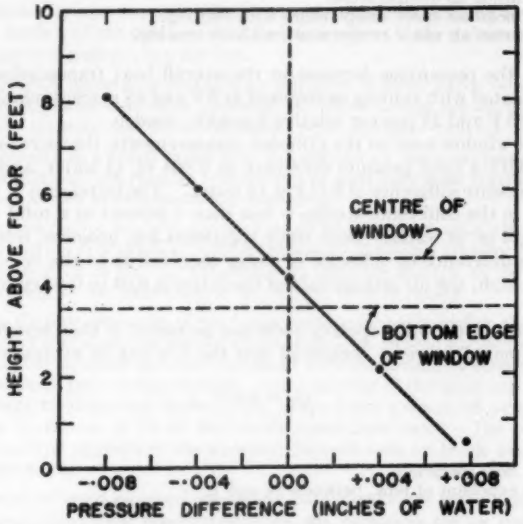

FIG. 9....PRESSURE DIFFERENCE BETWEEN WARM AND COLD ROOMS *vs.* HEIGHT

It is clear that cold room studies of condensation between the panes of double windows are of little value unless the total pressure differences across the window are measured. Even then, the observations are of use only in confirming some other approach for determining condensation performance, unless the total pressure differences correspond to those for which the window is being considered.

APPLICATION OF VENTING TO WINDOW CONDENSATION CONTROL

It has been shown that venting of the air space to outside through chimney action is an effective means of controlling condensation between the panes of double windows. It will be recognized, however, that excessive venting will lower the mean temperature of the air space and increase overall heat transmission. Taking the air space as equivalent to 2 surface conductances its mean temperature can be

estimated by equating the heat flow through the inner pane to the heat flow through the outer pane plus that carried away by the ventilating air. The apparent overall heat transmission coefficient with venting can then be expressed as

$$U' = U\left(\frac{T_i - T_o'}{T_i - T_o}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

where

U' = apparent overall heat transmission coefficient with venting.
U = overall heat transmission coefficient without venting.
T_i = inside air temperature.
T_o' = mean air space temperature with venting.
T_o = mean air space temperature without venting.

On this basis the percentage increase in the overall heat transmission coefficient has been calculated with venting as required at 0 F and 85 percent relative humidity outside, and 70 F and 25 percent relative humidity inside.

For the test window used in the diffusion measurements, the increase in U value is 1 percent with a total pressure difference of 0.004 in. of water, and 2.5 percent with a total pressure difference of 0.014 in. of water. The increase in U value for the window used in the cold room studies is less than 1 percent at a total pressure difference of 0.014 in. of water. Both these specimens are, however, relatively tight. At a pressure difference of 0.301 in. of water (equivalent to the velocity pressure of wind at 25 mph) the air leakage rate of the latter is 0.04 to 0.05 cu ft per min per ft of crack.

Leakage through the other specimen was not measured at this pressure difference. It can be estimated, however, assuming that the flow can be expressed as

$$Q_l = c(h)^n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

where

c = flow coefficient.
n = exponent of flow, between ½ and 1.

The exponent n is about 0.55 for the test units. The flow through the specimen used in the diffusion tests is then 0.07 to 0.08 cu ft per min per ft of crack at a pressure difference of 0.301 in. of water.

These leakage rates are probably representative of the leakage through cracks around the inner pane of many residential and commercial windows of the casement, awning, or hopper types, having inner and outer glazing in the same sash. There are other window types having a much lower resistance to flow from inside to the air space. For example, the Specification of the Aluminum Window Manufacturers Association permits a maximum air leakage rate of 0.5 cu ft per min per ft of crack at a pressure difference of 0.301 in. of water for a double double-hung window.[3]

An approximation of the venting requirements and the resulting increase in U value for such a window, for the conditions just referred to, can be made by extrapolating flow rates on the basis of Equation 15. If it is assumed that all the resistance to flow is from inside to the air space, the increase in U value is about 9 percent at a pressure difference of 0.004 in. of water and about 16 percent at a pressure difference of 0.014 in. of water. If it is assumed that the resistance to flow is normally evenly divided between inner and outer sash the increases in U value are 13 percent and

21 percent respectively with the required venting. The advantage of a tight inner sash is apparent.

In applying principles of venting to the design of a window, ambient conditions should be carefully selected. It is usually not necessary to design for the most extreme condition, since some condensation can generally be tolerated and time is required for its accumulation. Since information on the actual total pressure differences across building walls is quite limited, application of venting principles should be combined with directed observations of field performance.

CONCLUSIONS

1. With outside temperatures lower than inside, condensation will ultimately occur on the inside surface of the outer pane of double windows if the gain in water vapor to the window space is greater than the loss.

2. Vapor pressure differences available for diffusion of water vapor into the air spaces are several times greater than those available for diffusion from the air space to outside, the ratio of the pressure differences increasing with decreasing outside temperature.

3. Vapor transfer as a result of air flow will depend on the total pressure difference across the window and the relative resistance and distribution of cracks around inner and outer panes as well as psychrometric conditions.

4. For windows having air leakage rates equal to or greater than the test windows (0.05 cu ft per min per ft of crack at 0.301 in. of water), the vapor transfer by air flow will probably be several times that by diffusion even with the window at the neutral zone, the ratio increasing as the height above the neutral zone increases.

5. With the air flow resistance of the inside pane sufficiently higher than that of the outside pane, and with openings around the outside pane located at the top and bottom of the air space, the air space can interchange air with the outside by *chimney action* even with inside pressures greater than outside. Such venting of the space can be used effectively to prevent condensation between the panes if the amount of outside air interchange is large in relation to the air flow to the space from inside. The ratio of outside to inside air required to prevent condensation depends only on inside and outside psychrometric conditions.

6. The number of vents required to prevent condensation for given design conditions can be predicted from simple pressure relationships, providing the air flow characteristics of the vents and the cracks around the inner pane are known.

7. Excessive air interchange between the air space and outside can result in significant increases in the overall heat transmission coefficient. It may therefore be impractical to apply venting to windows having high leakage rates.

8. There is a lack of detailed information on actual conditions in buildings that affect condensation on the inside surface of the outer pane. In particular, further information on total air pressure differences between inside and outside, and inside relative humidities, is desirable.

9. Unless conditions of test, including total air pressure differences, are similar to those expected in the field the results of observations of condensation on the outer pane in cold room studies are not directly applicable and are of doubtful value except for research purposes.

ACKNOWLEDGMENT

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assistant, who assembled the equipment and took most of the records. The advice of N. B. Hutcheon, Assistant Director of the Division of Building Research is gratefully acknowledged.

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DISCUSSION

R. W. MCKINLEY, Pittsburgh, Pa.: The authors have added much useful information regarding vapor transmission and condensation characteristics of double windows. They have indicated that the indoor edge seal is especially important since it can importantly influence the thermal insulating value of the window. Usually, insulation is the primary justification for double glass.

Since most windows are included in the building so that one may see in or out, they are subject to critical visual examination. Therefore, it is important either to provide for regular, convenient cleaning of all glass surfaces or to design the double windows so that cleaning of inner surfaces is not necessary. This is another important reason for a reliable interior edge seal and for reducing condensation to a minimum. Designs should either prevent breathing *completely* or should filter dust from the entering air (i.e. the air entering the space between the panes.) If a unit is to breathe, it should breathe outdoor air only.

Unsightly and irregular accumulations of dirt on the interior surfaces of the glass can and should be avoided.

1686

No. 1686

EVAPORATIVE COOLING FOR COMMON STORAGES OF FRUITS AND VEGETABLES

By R. S. Ash*, Phoenix, Ariz.

STORAGE of fall and winter fruits and vegetables offers a practical method of minimizing price fluctuations in the market place. Ideally, produce should be packed and shipped immediately after being harvested. However, a large portion of fall and winter crops are held in storage to provide a regulated supply to the market throughout the winter and spring.

When fruits and vegetables are harvested they continue to behave as living organisms. Like all other living things they carry on the process called respiration. The more rapidly it takes place, the more quickly the produce will deteriorate. Since respiration varies with the temperature, lowering the temperature results in a slower rate of ripening and food materials, flavor, fresh appearance and general quality are conserved.

If fruits and vegetables are put in common or air-cooled storages (insulated buildings cooled during the fall months by introducing night air), the temperature of the produce in storage is usually close to the prevailing mean outside temperature[4, 8, 13, 21, 23, 26]. During the months of September and October when there are periods of high temperature the produce ripens prematurely. It also wilts and shrivels from becoming too dry. (A 40-lb bushel of apples can lose as much as 2 to 4 lb, and this loss of weight represents water evaporated from the fruit[23].) If little fresh air is introduced into the storage, carbon dioxide and ripening gases which evolve from the respiration process will not be carried away. Another problem in the case of apples may be the presence of esters which will cause storage scald. There may also be the problem of odors absorbed from the storage room.[29] Although the picture for apples has been described, the storage problem of other fruits and vegetables is similar.

These common storages for fruits and vegetables can be improved considerably through the use of the simple principle of evaporative cooling, a form of air condi-

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* Assistant to the President, International Metal Products Co. Member of ASHRAE.

[4] Exponent numerals refer to References.

Presented at the Annual Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Lake Placid, N. Y., June 1959.

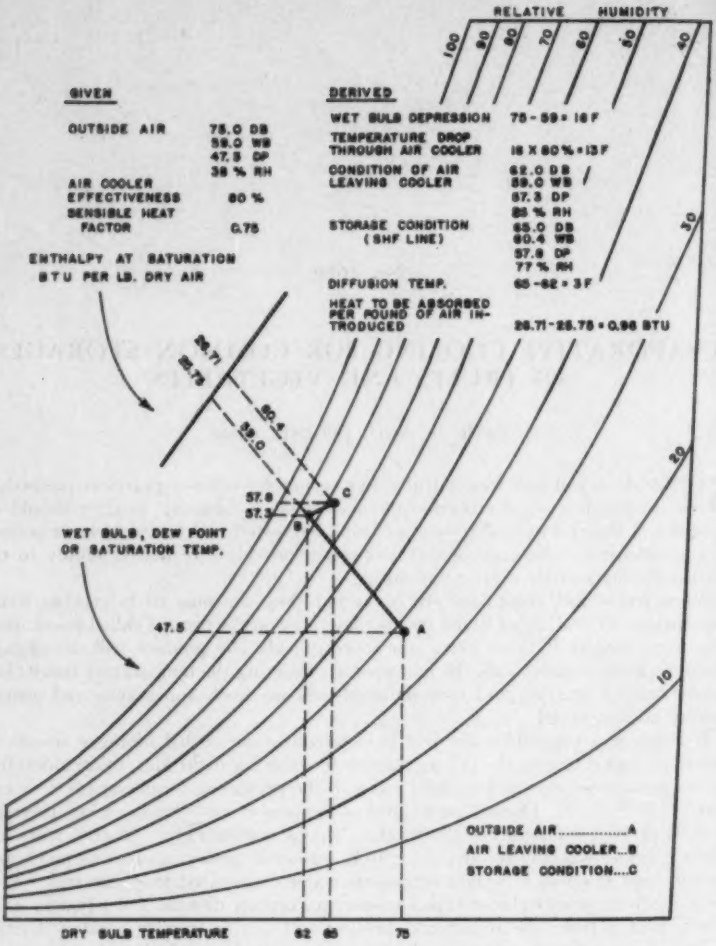

FIG. 1....PSYCHROMETRIC CHART INDICATES A TYPICAL EVAPORATIVE COOLING PROBLEM

tioning based upon the evaporation of water. Evaporative cooling involves an adiabatic exchange of heat. The sensible heat of the air is reduced proportionally to the amount of evaporation that takes place. The water assumes the wet-bulb temperature of the air and cooling proceeds, with the enthalpy or total heat content of the air remaining essentially constant. Humidification occurs as a result of the vapor pressure exerted by the water, which is higher than that corresponding to

the entering air dew-point. In the process, the dew-point temperature rises and the dry-bulb temperature falls, with the wet-bulb temperature remaining constant. The maximum possible dry-bulb temperature reduction is the difference between the entering air dry-bulb and the wet-bulb temperatures, and this is usually referred to as the wet-bulb depression. Were it possible to cool the air an amount equal to the wet-bulb depression, the air would be completely saturated. Since this is not the case in actual practice, evaporative cooling systems operate at something less than 100 percent effectiveness.

In this connection, the temperature of the air resulting from the evaporative cooling process should never be considered as the storage temperature. The latter will be several degrees higher. The temperature difference between the storage-room temperature and the evaporative cooler discharge temperature is usually called the diffusion temperature. It depends upon the heat load and the amount of cool air discharged into the storage. In calculating the heat load no attention need be paid to the latent load of respired water by the commodity, as the system operates on the basis of 100 percent fresh air and any moisture pick up is discharged to the outside. Furthermore, in common storages for fruits and vegetables any increase in relative humidity is desirable.[8] Fig. 1 is a skeleton psychrometric chart on which is illustrated a typical evaporative cooling problem involving a heat load that is characteristic of a potato storage house during the early fall, when it is desirable to keep a potato temperature of about 60 F to encourage wound healing.

APPLES

Apples must be harvested at the correct stage of maturity and then be stored immediately for best results.[11, 26] When the nights are cool, apples picked in the late afternoon can be left in the orchard over night to lose some of their field heat. To minimize loss of moisture, both the apples and the field crates in which they are stored should be wetted down at time of storage. An air space of 4 in. or more should be left along each wall, and stacking should not be permitted higher than the bottom of the cooling system ducts. The crates should be so stacked that air may circulate between them. A few inches of space left between each tier insures a more rapid air movement through the stacks. Lines painted on the floor of the storage room to indicate the spaces for the rows of boxes are helpful in storing the fruit.

Temperature in the storage should be reduced as rapidly as possible to the ideal apple storage temperature of 31 to 32 F.[13, 21, 22, 33] This can usually be attained by the middle of November, as the daily minimum temperature in most storage areas is below this level by that time of the year. High relative humidity is also of vital importance in keeping apples in prime condition. A relative humidity of from at least 85 to 90 percent should be maintained. This is the approximate water content of apples, and the pull on the fruit for moisture or the evaporative power of the air will be slight and shrinkage of the fruit will be correspondingly lessened.[11] If the temperature is above 36 F such as when the storage is being filled in September and October, the relative humidity should be from 90 to 95 percent. An atmosphere of 95 percent or higher is necessary to entirely prevent moisture loss. Humidity as high as this, however, favors mold growth on the fruit, crates and walls of the storage.[29]

Evaporative cooling systems for apple storages should be designed to distribute the cool air to all parts of the storage. The evaporative coolers may be floor mounted, or located near the ceiling in a fan room. The system should be designed

to discharge the air horizontally at the ceiling level. The maximum practical air throw is about 30 ft. In wide storages where the width exceeds 60 ft, it is usually desirable to install 2 trunk ducts so spaced that the distance to the wall does not exceed the maximum throw. Canvas hung at strategic positions will force the air to travel through the fruit as it moves to the exhaust opening. The space between the outer stacks and the wall should be blocked at the ends when the storage room is not filled. Stacking crates at the end of the space during loading is enough. If the ends are not blocked, the air tends to move out the ends and around the stacks rather than through them.

The evaporative cooler should be sized to give a 3-minute air change for quick and even cooling. The total volume of the storage should be considered in calculating the size of the system. The heat load on the evaporative cooling system is sizeable as it consists not only of the heat leakage into the storage, but the field heat

TABLE 1. . . . EVOLUTION OF HEAT OF APPLES AT VARIOUS TEMPERATURES

TEMPERATURE, F	HEAT PER TON IN 24 HR, BTU
32	700
35	1000
40	1500
45	2200
50	3500
55	4600
60	5800
65	6900
70	8000
75	9000
80	10000
85	11000

of apples as well as the heat of respiration. The heat leakage into the apple storage can be calculated like any other building. Field heat removal must be considered as the apples are placed in storage at outdoor temperature.[25] (If the apples have been left in the sun the temperature may be even higher.) Assuming the apples are placed in storage at outside temperature they must be cooled to the prevailing storage room temperature. The specific heat of apples above freezing is 0.87 Btu. A weight of 50 lb per bushel is assumed for apples. This includes the weight of the field crate which also must be cooled. It then requires 50 × 0.87 or 43.5 Btu to cool a bushel of apples 1 F deg. To cool 1000 bushels 10 F deg, 43.5 × 1000 × 10 or 435,000 Btu are required. Since this is for 24 hr, the hourly requirements are 18,125 Btu. In addition to the heat leakage into the storage and the field heat of the apples, it is necessary to take into consideration the heat of respiration.

The approximate evolution of heat through the respiration of apples at different temperatures is shown in Table 1.[24, 29] These figures represent total heat of respiration and therefore include transpiration. In the process of respiration the water that is evolved is contained within the cellular interstices of the fruit, and is released only as a direct result of transpiration.[5] About 25 percent of the heat of evolution shown in Table 1 represents transpiration losses, and can be ignored in calculating the load imposed on the cooling system by the respiration process.[21, 22]

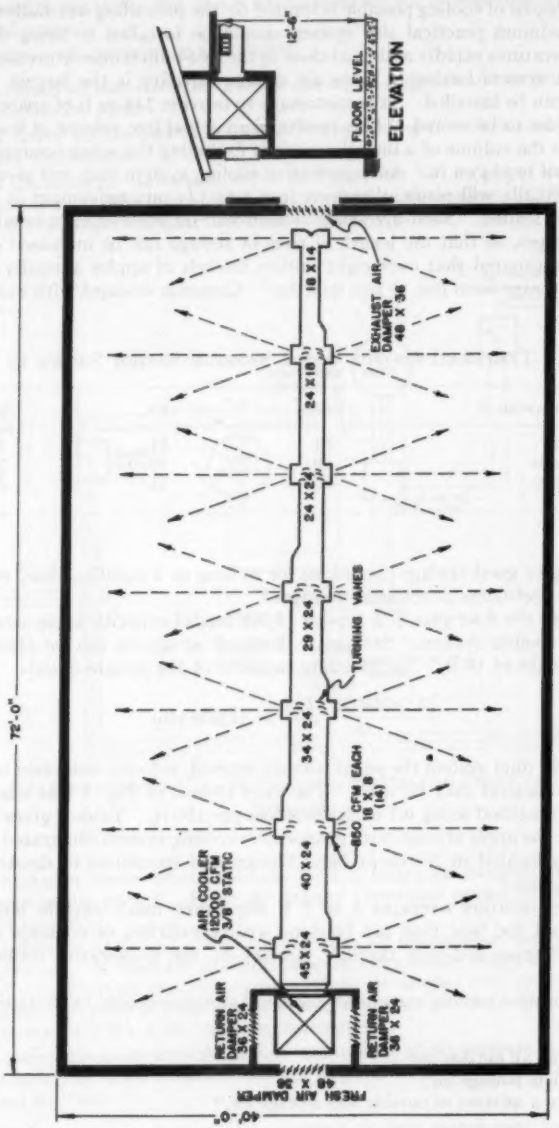

FIG. 2. . . .FLOOR PLAN—APPLE STORAGE, 15,000 BUSHEL CAPACITY

Since the degree of cooling possible is limited by the prevailing wet-bulb temperature, the maximum practical size system should be installed to bring down the storage temperatures rapidly and to as close to the wet-bulb temperature as possible. In general, a system having a 3-min air change capacity is the largest practical system that can be installed. It is customary to provide 2½ cu ft of space for each bushel of apples to be stored. This results in an actual free volume of less than 50 percent, since the volume of a bushel container excluding the space occupied by the container itself is 1¼ cu ft. An evaporative cooling system that will give a 3-min air change actually will result, therefore, in a 1 to 1½ min movement of air when the storage is loaded. Such a cooling system will improve storage conditions in common storages, so that the length of time in storage can be increased considerably. It is estimated that over eight million bushels of apples annually are held in common storage from one to two months.[7] Common storages with evaporative

TABLE 2. . . . TEMPERATURES FOR APPLE STORAGE SYSTEM SHOWN IN FIG. 2

TEMPERATURE, F	SEPT.	OCT.	NOV.
Mean Outside	61	51	39
Mean Wet-Bulb	49	39	29
Average Storage	55	45	32

cooling will give good storage conditions for as long as 4 months, thus, more than doubling the usefulness of common storages.[9]

Fig. 2 shows the floor plan of a typical 15,000 bushel capacity apple storage with evaporative cooling system. Storage is designed so apples can be stacked to a maximum height of 10 ft. Air handling capacity of the system equals

$$\frac{72 \times 40 \times 12\frac{1}{2}}{3} = 12{,}000 \text{ cfm}$$

In sizing the duct system the equal friction method, velocity reduction method or static regain method may be used. The duct system in Fig. 2 was sized by the equal friction method using 0.1 in. friction loss per 100 ft. Table 2 gives temperature data for the apple storage with evaporative cooling system illustrated in Fig. 2. This plant is located in Northern New Mexico and specializes in dessert quality fruit for gift pack.[9]

Storage temperature averages 6 to 7 F deg below mean outside temperature which is about the best that can be done with ventilation of common storages.[4] Fruit is kept crisp and firm through January by the evaporative cooling of the storage.[9]

The evaporative cooling system for common storages should be designed so that it is possible:

1. To supply all outside air;
2. Recirculate storage air;
3. Circulate a mixture of outside and storage air.[31]

The setting of the dampers depends upon the outside and inside temperatures. The setting of the outside air, return air and exhaust dampers may be done manually

or by automatic control. Fig. 3 is a wiring diagram for modulating or potentiometer type of control.

The sequence of operation is as follows:

1. No. 1 return air thermostat set at 32 F starts and stops fan and recirculating water pump of evaporative cooler.

2. No. 4 freeze up thermostat set at 35 F stops pump if recirculated sump water drops to this temperature.

3. No. 2 discharge air thermostat set at 30 F operates No. 5 damper motor to open fresh air damper on a rise in temperature and closes return air damper. Fresh air damper closes on a fall in temperature with return air damper opening.

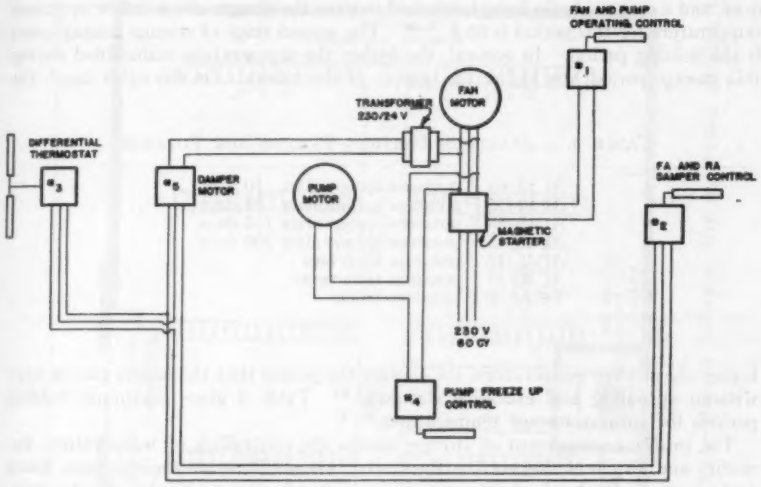

FIG. 3. . . .EVAPORATIVE AIR COOLER CONTROL SYSTEM FOR FRUIT AND VEGETABLE STORAGE

4. No. 5 damper motor modulates fresh air and return air dampers at direction of No. 2 proportioning control. Fresh air damper closes and return air damper opens when fan is shut off.

5. No. 3 differential thermostat compares return air temperature with fresh air wet-bulb temperature. If fresh air wet-bulb temperature exceeds return air temperature, control will close fresh air damper and open return air damper until such time as fresh air wet-bulb temperature is below return air temperature. No. 5 damper motor is then returned to control of No. 2 discharge air thermostat.

6. Automatic shutter or pressure damper opens under static pressure in storage room to exhaust spent air, when system introduces fresh air at time fresh air damper modulates in open position.

In addition to these controls a cycle repeater or time switch may be installed in the system to provide intermittent fan operation when no cooling is required but

air circulation is desired. If additional humidity should be needed, spray nozzles can be installed in front of the outlet grilles to open when the fan of the evaporative cooler is in operation. The water lines should be vented so that they will drain free and not freeze up.

POTATOES

Potatoes harvested from late August through November are usually put in storage and may be held for as long as 6 to 8 months.[6, 15] The storage period of potatoes is divided into 2 parts: the wound-healing or curing period and the holding period. The wound-healing period immediately follows harvest. During this period the bruises and other wounds caused during the harvesting operation heal over, and a corky suberin layer forms and reduces the danger of rot. The optimum temperature for this period is 60 F.[10, 30] The second stage of storage management is the holding period. In general, the higher the temperature maintained during this storage period, the higher the quality of the tubers. On the other hand, the

TABLE 3. . . .MAXIMUM HOLDING PERIODS FOR POTATOES

At 55–60 F	potatoes sprout after 70 days
At 50–55 F	potatoes sprout after 89 days
At 45–50 F	potatoes sprout after 126 days
At 40–45 F	potatoes sprout after 200 days
At 35–40 F	potatoes keep best
At 30–35 F	potatoes turn sweet
Below 30 F	potatoes freeze

higher the storage temperature, the shorter the period that the tubers can be kept without sprouting and excessive shrinkage.[28] Table 3 gives maximum holding periods for various storage temperatures.[17, 27]

The proper management of storage means the controlling of temperature, humidity and air circulation so that the stored tubers will retain their texture, flavor and maximum food value with a minimum of loss from rot, shrinkage and sprouting. Potatoes may give off as much as 175 Btu per hundredweight each 24 hr when stored at high temperatures.[12] Lower storage temperatures result in as little as 20 Btu. Temperature of long term storage should be approximately 38 F.[2, 18] A lower temperature causes the starch to turn to sugar, while a higher temperature results in excessive shrinkage through accelerated respiration and transpiration.[2]

Humidity of the air immediately around the potatoes should be as high as possible. In practice, 85 to 90 percent relative humidity should be maintained.[18] In the fall at harvest time, large volumes of cool air should be drawn through the storage to bring the potatoes to proper storage temperature. After a temperature of 38 to 40 F is reached, cool moist air is needed to prevent the temperature from rising above that point and to minimize shrinkage.[5, 24] During the winter when cooling is not needed, proper air circulation is still required to maintain uniform temperature and humidity. This prevents freezing near the walls and sprouting or spoilage by a too high temperature at the bin center.

Evaporative cooling systems for potato storage houses should be designed for through-cooling with the air passing directly through and in contact with the potato pile.[6] Capacity of cooling system should be based on a 3-min air change to give

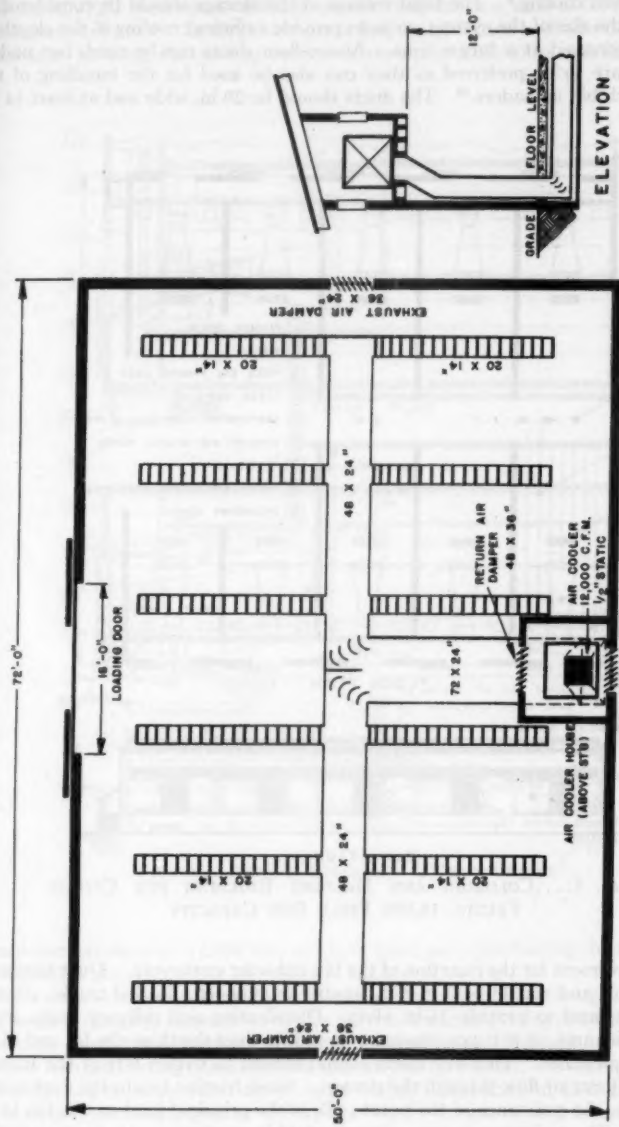

FIG. 4. . . .FLOOR PLAN—POTATO STORAGE HOUSE, 25,000 BUSHEL CAPACITY

quick and even cooling.[3] The total volume of the storage should be considered in calculating the size of the system, so as to provide sufficient cooling if the depth of storage is increased at a future time. Above-floor ducts can be used, but under-floor ducts are to be preferred as they can also be used for the handling of the potatoes with bin unloaders.[15] The ducts should be 20 in. wide and at least 14 in.

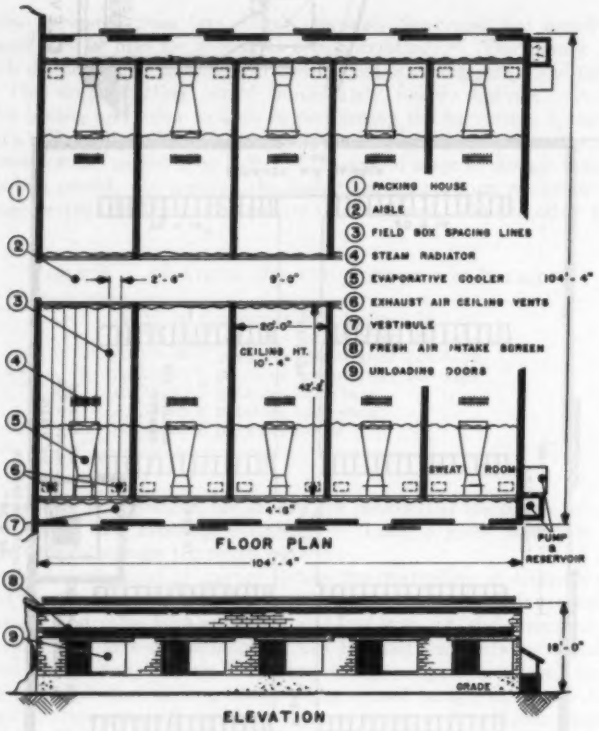

FIG. 5. . . .COLORING AND HOLDING BUILDING FOR CITRUS FRUITS, 16,000 FIELD BOX CAPACITY

deep to allow room for the insertion of the bin unloader conveyors. Duct tops must be removable and should be 3- x 8-in. lumber to support a loaded truck. Boards should be spaced to provide ½-in. slots. Distributing and delivery ducts act as extended plenums, so it is not necessary to reduce duct depth as the far end of the ducts is approached. Delivery ducts should extend to within 6 ft of the walls to provide uniform air flow through the storage. Since friction loss in the duct system is negligible, the resistance of the potato pile is the principal load on the fan in the air cooler. The resistance is approximately 0.25 in. static pressure (potato storage should not exceed 12 ft depth). The additional resistance of the ducts, inlets,

dampers and air cooler pads will add another 0.25 in. or a total of 0.50 in. that the fan must overcome.[34] In large storages it may be advisable to divide the house into 2 sections to keep the cooling system within practical limits. A single system

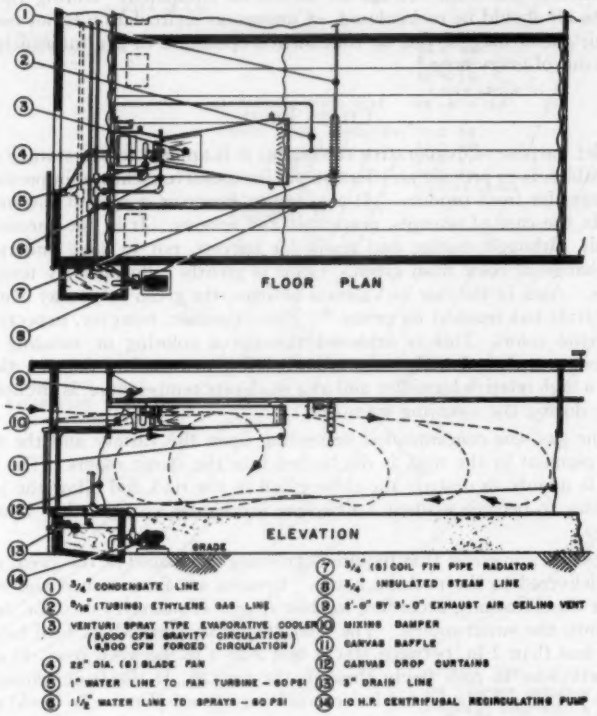
FLOOR PLAN
ELEVATION
GRADE
① 3/4" CONDENSATE LINE
② 5/16" (TUBING) ETHYLENE GAS LINE
③ VENTURI SPRAY TYPE EVAPORATIVE COOLER (6,000 CFM GRAVITY CIRCULATION) (8,800 CFM FORCED CIRCULATION)
④ 22" DIA. (8) BLADE FAN
⑤ 1" WATER LINE TO FAN TURBINE – 60 PSI
⑥ 1 1/4" WATER LINE TO SPRAYS – 60 PSI
⑦ 3/4" (8) COIL FIN PIPE RADIATOR
⑧ 3/4" INSULATED STEAM LINE
⑨ 2'-0" BY 2'-0" EXHAUST AIR CEILING VENT
⑩ MIXING DAMPER
⑪ CANVAS BAFFLE
⑫ CANVAS DROP CURTAINS
⑬ 3" DRAIN LINE
⑭ 10 H.P. CENTRIFUGAL RECIRCULATING PUMP

FIG. 6. . .SWEAT ROOM FOR CITRUS FRUITS

should not handle over 12,000 cfm of air, if main and distributing ducts are to be of reasonable size.

Fig. 4 shows the floor plan of a potato storage house located on Long Island, N. Y. The house is designed so potatoes can be piled to a maximum depth of 10 ft. Ducts have been sized by the velocity reduction method using 1000 fpm for trunk duct, 750 fpm for distributing ducts and 500 fpm for delivery ducts. Duct system is laid out so that all delivery ducts are of equal length and spaced on 12-ft centers.

The operation of evaporative cooling systems for potato storage houses is similar to that for apple storages. All outside air is normally used during the cooling period. As an example, assume that the desired temperature within the storage is to

be 40 F. The system should operate on 100 percent outside air, if the temperature of the air discharged from the cooler is as cool or cooler than the inside of the storage, and the storage temperature is not lower than 40 F. Should the temperature of the air discharged from the cooler become warmer than the air inside the storage, the air within the house should be recirculated. If the outside air drops below 40 F, some outside and some storage air should be circulated. During the winter months the air should be recirculated. Continuous recirculation is not necessary, and the air cooler may be put on time switch operation to run intermittently 10 to 15 min out of every hour.[6]

CITRUS FRUIT

The chief purpose of evaporative cooling, as it is applied to the storage of fruits and vegetables, is to provide an effective yet inexpensive means of improving common storages for fresh produce. It also serves, however, a special and important function in the case of oranges, grapefruit and lemons. It is not uncommon for citrus fruit, although mature and ready for harvest, not to have undergone the natural change in color from green. Color is greatly influenced by temperature variations. And, in the case of Valencia oranges, the green color may even return after the fruit has reached its prime.[19] The consumer, however, expects fruit of characteristic color. This is achieved through a coloring or *sweating* process. Special rooms equipped with evaporative cooling are used to degreen the fruit.[1] Air, with a high relative humidity and at a moderate temperature, is circulated continuously during the sweating operation.

Ethylene gas, the concentration depending upon the variety and the intensity of green pigment in the rind, is discharged into the sweat rooms. The effect of ethylene is mainly to destroy the chlorophyll in the rind and allow the yellow or orange color to become evident. Ethylene accelerates changes that would occur naturally.

Oranges and grapefruit that require degreening are placed in the sweat rooms as soon as delivered to the packing house. Lemons are first washed and separated into color classifications, according to their degree of maturity or color, before being put into the sweat rooms. The fruit should be stacked six field boxes high with not less than 2 in. between stacks and 3 to 4 in. between rows, to allow the air and ethylene to pass freely through the fruit.[19] If the fruit temperature is above 75 F, it should be cooled before applying the ethylene. To avoid excessive shrinkage and other ill effects during the cooling down period, the moisture content in the sweat room must be maintained at a high level. Water should be sprayed onto the fruit if the fruit temperature is extremely high, near 100 F. Ventilation should be continuous during the cooling down period.

In Fig. 5 is depicted the coloring and holding building adjoining a packing house in Phoenix, Ariz. Fig. 6 is a detailed drawing of one of the 10 sweat rooms in the coloring building. Each room is approximately 20 × 40 ft with a 10-ft ceiling, designed to accommodate 1600 field boxes each containing 50 lb of fruit. A temperature of 70 F is maintained in the sweat room during the degreening operation with a relative humidity of 88 to 90 percent. (In the gulf states 80 to 85 F temperature with 90 to 92 percent relative humidity is recommended.[14, 32]) The evaporative cooler is designed to deliver 4 cfm per field box or approximately 6500 cfm at maximum load. 5000 cfm can be obtained with gravity circulation through the induced or venturi action of the spray nozzles in the throat of the cooler.[20]

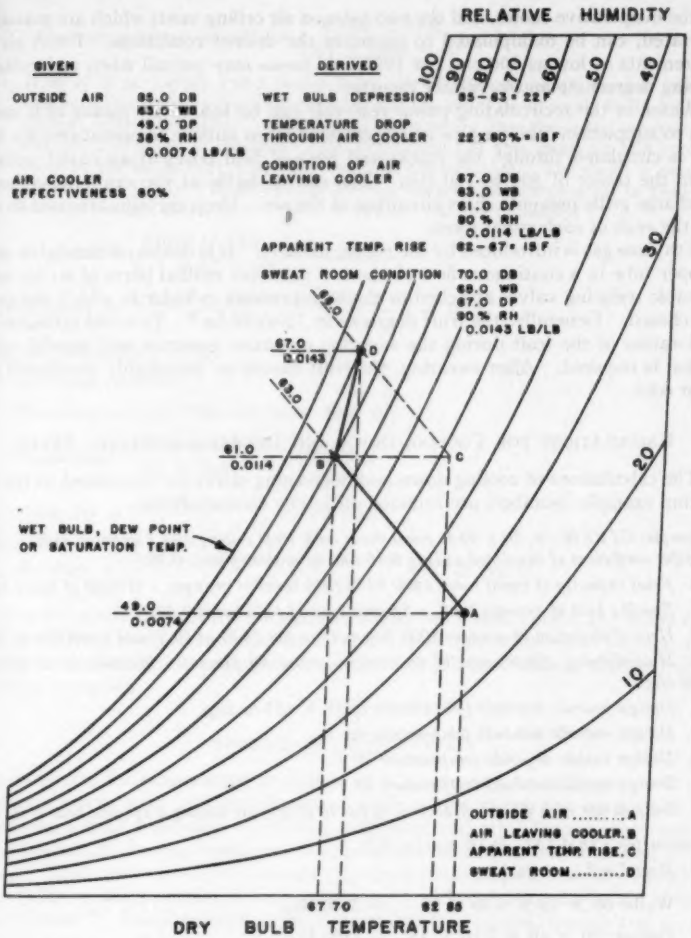

FIG. 7....CHART ILLUSTRATES CITRUS SWEAT ROOM CONDITIONS

Additional air capacity is available from the water-turbine-driven fan if needed (fan alone will deliver 3600 cfm). Each of the 6 spray nozzles will deliver approximately 4 gph maximum at 60 psi water pressure. High relative humidity is required to minimize shrinkage of the fruit. Although the cooler delivers air at a practically saturated condition, additional free moisture from the spray nozzles is available if necessary to increase the cooling effect, and thereby keep the temperature and humidity in the sweat room within design conditions. The mixing damper

in the evaporative cooler, and the two exhaust air ceiling vents which are manually operated, can be manipulated to maintain the desired conditions. Fresh air requirements as low as 300 cfm per 1000 field boxes may prevail when recirculating during degreening and/or winter months.

Water in the recirculating pump reservoir can be heated by means of a steam coil to supplement the fin pipe steam radiator when outside temperatures are low. Air is circulated through the stacks and rows of field boxes at an outlet velocity from the cooler of 500 to 750 fpm. The canvas baffle at the evaporative cooler discharge grille prevents short circuiting of the air. Drop curtains are used to seal off the ends of each sweat room.

Ethylene gas is introduced by the trickle method. It is delivered through a small copper tube in a continuous flow of about 5 parts per million parts of air by using suitable reducing valves attached to the high-pressure cylinder in which the gas is purchased. Generally, the fruit degreens in 72 to 96 hr.[20] To avoid excessive deterioration of the fruit during the sweating operation constant and careful supervision is required. After sweating, the fruit should be thoroughly ventilated and kept cool.

CALCULATIONS FOR COOLING-DOWN AND DEGREENING CITRUS FRUITS

The calculations of cooling down and degreening citrus are illustrated in the following example (numbers are rounded off in the computations).

Example: GIVEN—a. 20 x 40-ft sweat room with 10-ft ceiling and 2 exposed walls. Heat transfer coefficient of insulated ceiling 0.10 and of masonry wall 0.30.

b. Fruit capacity of sweat room 1600 50-lb field boxes of oranges. Weight of boxes 12 lb.

c. Specific heat of oranges 0.90. Specific heat of field boxes 0.40.

d. Heat of evolution of oranges 8000 Btu per ton per 24 hr at 80 F and 6500 Btu at 70 F.

e. Humidifying effectiveness of evaporative cooler 80 percent. Maximum air delivery 6500 cfm.

f. Design outside dry-bulb temperature of 85 F. (24 hr avg)

g. Design outside wet-bulb temperature 63 F.

h. Design inside dry-bulb temperature 70 F.

i. Design inside wet-bulb temperature 68 F.

j. Outside air with weight of 13.9 cu ft per lb of dry air having a specific heat of 0.24.

Solution for COOLING DOWN—24 HR:

1. *Heat Load (sensible)*

Walls:	$60 \times 10 \times 0.30 \times 15$	= 2700 Btu
Ceiling:	$20 \times 40 \times 0.10 \times 15$	= 1200 Btu
Fruit:	$\frac{1600 \times 50 \times 0.90 \times 15}{24}$	= 45000 Btu
Boxes:	$\frac{1600 \times 12 \times 0.40 \times 15}{24}$	= 4800 Btu
Respiration:	$\frac{80000 \times 8000}{24 \times 2000}$	= 13300 Btu
Total		67000

2. *Cooler outlet temp* = $85 - [(85 - 63) \times 0.80] = 67$ F

3. *Minimum temp rise* possible $= \dfrac{67000 \times 13.9}{0.24 \times 6500 \times 60} = 10$ F

4. *With 67 F air supply* and a temperature rise of 10 F using 100 percent fresh air maximum air delivery the final conditions in the sweat room would be 77 F and 58 percent relative humidity. Additional free moisture from the spray nozzles in the evaporative cooler is required. The evaporation of this moisture will cool the room by the latent heat required to vaporize the moisture. The skeleton psychrometric chart in Fig. 7 illustrates the process whereby the temperature in the sweat room is brought down to 70 F with 90 percent relative humidity.

5. *Fresh air* $= \dfrac{67000 \times 13.9}{0.24 \times 15 \times 60} = 4300$ cfm

6. *Moisture* $= \dfrac{(0.0143 - 0.0114) \times 4300 \times 60}{13.9} = 50$ lb

Solution for DEGREENING:

1. *Heat load (sensible)*

Walls and ceiling: $2700 + 1200 = 3900$ Btu

Respiration: $\dfrac{80000 \times 6500}{24 \times 2000} = 10800$ Btu

2. *Fresh Air* $= \dfrac{14700 \times 13.9}{0.24 \times 15 \times 60} = 950$ cfm

3. *Moisture* $= \dfrac{(0.0143 - 0.0144) \times 950 \times 60}{13.9} = 12$ lb per hr

Since the air delivery of the evaporative cooler using both sprays and fan is 6500 cfm, the mixing damper and exhaust air vents should be adjusted to provide 4300 cfm fresh air during the cooling down period, and 950 cfm fresh air during the degreening operation.

STORAGE OF CITRUS FRUITS

Another use for evaporative cooling is as a supplement to refrigeration in the storage of citrus fruit. Although oranges have a long harvesting season and are considered a fresh fruit crop, marketing conditions and other factors may make it advisable to hold them in temporary storage at the packing house for as long as one month. And in the case of lemons and grapefruit these crops are usually stored for some time. A large portion of the former crop is picked during the period of least consumption.[16] Recommended storage conditions for oranges, lemons and grapefruit are as follows.[20]

	ORANGES	LEMONS	GRAPEFRUIT
Temperature	50	58	60
Relative humidity	88–90	84–88	86–88

While these conditions require refrigeration in the summer they can often be met with evaporative cooling during the fall, winter and spring when the outside wet-bulb temperature is low.

SUMMARY

Large quantities of fruits and vegetables must be stored or there would be market gluts in the fall at harvest time. Furthermore, good marketing demands a regulated supply throughout the whole selling period. Farm storages for such fruits and vegetables may be classified as common storage and refrigerated storage. Common storage can be improved considerably through the use of the simple principle of evaporative cooling. High humidity to prevent shrinkage and lower temperature to reduce respiration are achieved with the additional benefit of fresh air to carry away ripening gases and to prevent odor.

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DISCUSSION

R. L. SUMMERHAYS, Rochester, N. Y.: What is the average relative humidity and wet-bulb temperature in the Long Island area of New York state? The author mentioned potato storage at Suffolk.

W. A. DANIELSON, Memphis, Tenn.: How is the air introduced through the floor? It would seem that there is a great chance of getting dirt down into the ducts and that it would be necessry to clean them out frequently.

W. T. PENTZER, Washington, D. C.: There are two questions. One is—was anything done to do more than reach the wet-bulb temperature, as for example, with apples early in the season when it is desirable to have them down in a day or two? The second question is—was there any attempt to add refrigeration or to take a stepwise reduction of air temperature? The one question is aimed to learn whether it might be advan-

tageous to use a little cold storage early rather than later when outdoor temperatures are sufficiently low to produce the temperature needed without refrigeration.

The other question arose from the mention that the mean temperature was reached. The use of common storage for potatoes is extremely important in Holland from information gathered at the Potato Institute. There it was indicated that before practicing above-ground storage where air from outside is used under controlled ventilation,—and which idea came from the U.S.A.—, they stored in pits in the ground, and the mean potato temperature was about the mean maximum daily temperature with the sun hitting the ground and warming it up, but with the introduction of controlled outside ventilation, not using wet-bulb and forced air up through the floor, the mean potato temperature approached very close to the minimum outside temperature. This would indicate a temperature a little better than mean.

W. T. SMITH, Washington, D. C.: Isn't it possible to get below the wet-bulb with two-stage evaporative cooling?

AUTHOR'S CLOSURE: From recollection, the daytime relative humidity in the Long Island area is 55 to 60 percent during the fall of the year and at night it can run up as high as 90 percent, although 70 to 75 percent is more characteristic of the nighttime relative humidity.

General Danielson's comment is correct. The potatoes are put into the storage in bulk and are often quite dirty. However, the tops of the ducts are removable for the insertion of the bin unloader and it is common practice to sweep out the ducts whenever they get so dirty that the flow of air is impeded.

With reference to the first question of Mr. Pentzer, if his suggestion could be accomplished it would further the use of evaporative cooling equipment, but unfortunately the wet-bulb temperature sets a limit and when conditions better than that are wanted it is best to go to cold storage and use refrigeration. It is very possible that it would be advantageous to use a little cold storage early rather than later.

What is being done overseas on potato storage, particularly in Holland as Mr. Pentzer states, is very interesting. An article recently appeared on the subject in one of the British heating and ventilating magazines. From information gathered by people at Cornell University and at the Long Island Vegetable Research Farm, which is an adjunct of Cornell, the author's understanding is that the mean temperature is about the best that can be achieved with mechanical ventilation. An evaporative cooling system will reduce storage temperature 6 to 7 F deg below the mean in most cases.

It is interesting to note that this use of evaporative cooling is an outgrowth of greenhouse evaporative cooling, of which the author has done quite a bit. A customer who had a small greenhouse was also in the potato storage business and the thought occurred to him that evaporative cooling would improve conditions in potato storages. That is how it started. Now, there are several hundred installations on Long Island, which is a place where one would scarcely expect to find evaporative cooling.

In reply to Mr. Smith's question, it is possible in theory to get all the way down to the dew-point temperature through staging. However, 2-stage cooling is usually not justified. It is doubtful whether it would be practical from an economic standpoint in the case of produce.

1687

No. 1687

CONTROLLED ATMOSPHERE APPLE STORAGE PROCESS AND ITS REQUIREMENTS ON REFRIGERATION STRUCTURES AND SYSTEMS†

By J. W. ZAHRADNIK*, AMHERST, MASS.

KEY to the controlled atmosphere process is maintaining a temperature of 37 F (± 1½ F deg); a relative humidity of 85–90 percent; and an atmosphere of 3 percent oxygen (O_2), 5 percent carbon dioxide (CO_2), and 92 percent nitrogen (N_2) and miscellaneous inert gases.[1,2] These storage conditions make it possible to preserve the quality of McIntosh apples for 6 to 8 months on a commercial basis.

After a controlled atmosphere (C-A) storage is properly loaded and sealed, the atmosphere composition is 17–21 percent O_2 and 79–83 percent N_2 and miscellaneous inert gases. Exact composition will depend on the amount of air exchange during loading. The Orsat apparatus is commonly used to determine atmosphere composition. This initial atmospheric composition must be modified to 3 percent O_2, 5 percent CO_2 and 92 percent N_2 through proper manipulation of the materials balance.

The apples continue respiring and that process requires O_2 and produces CO_2 simultaneously.[3] Thus, there is a reduction in O_2 level and an increase in CO_2 level. This is illustrated in Fig. 1. As this respiration continues the CO_2 level will reach 5 percent, one of the controlled conditions. Until this point is reached, in a relatively gas tight storage, the total materials balance has not been disturbed volumetrically; the materials have been changed in composition only. However, as respiration continues, the CO_2 level will rise above 5 percent and it is at this condition that CO_2 is removed by scrubbing the atmosphere with sodium hydroxide solution and the materials balance on a volumetric basis must be maintained by

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[1] Exponent numerals refer to References.

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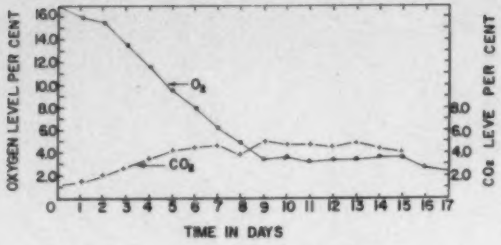


FIG. 1. . . . TIME *vs* OXYGEN LEVEL

the addition of N_2. In this way the CO_2 level is held relatively constant, the apples continue to bring down the O_2 level, and the materials balance is further maintained by the addition of N_2 until final process conditions are reached. These various atmosphere conditions are illustrated in Fig. 2.

When final process conditions are achieved, (after from 10–20 days of storage), air must be added to maintain the materials balance instead of N_2, since the apples

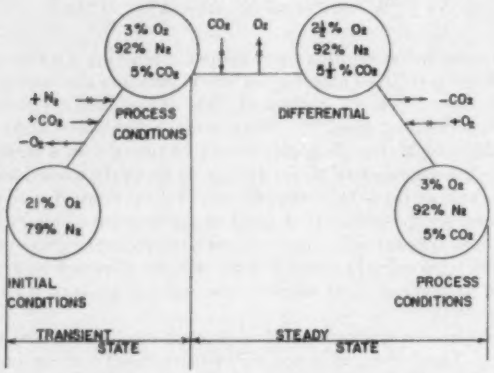
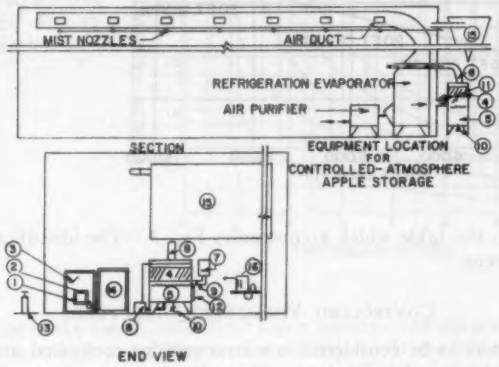


FIG. 2. . . . THE CONTROLLED ATMOSPHERE PROCESS

continue to respire (although at a much reduced rate), and require the O_2 present in air. The addition of air to maintain 3 percent O_2 and the scrubbing of CO_2 to maintain 5 percent continues throughout the storage season. These storage conditions are maintained at present by manual control, with some electric time clock programming of the CO_2 scrubber operation.

EQUIPMENT FUNCTION AND LOCATION

Equipment necessary to maintain process conditions is illustrated schematically in the diagram of Fig. 3. The purpose of each piece of specialized equipment is



No.	Name	Function
1	Access panel	To permit emergency entry for minor repairs and inspection.
2	Porthole	To permit withdrawal of fruit samples during storage season.
3	U safety trap	To prevent build up of excessive internal pressure or vacuum.
4	Coke packing	To provide surface area for contact between the CO_2 of atmosphere with the sodium hydroxide of the scrubber.
5	Sump	To provide storage for the sodium hydroxide solution of the scrubber.
6	Circulating pump	To pump the sodium hydroxide solution from the scrubber sump to the coke packing.
7	Mixing bucket	To provide a container for dissolving the sodium hydroxide flakes into water.
8	Blower	To circulate the storage room atmosphere through the CO_2 scrubber.
9	Valve	To fill the scrubber sump.
10	Valve	To drain the scrubber sump.
11	WB DB thermometer	To determine relative humidity.
12	Sight glass	To determine the level of the sodium hydroxide solution in the scrubber sump.
13	Nitrogen tank	To maintain the materials balance during the initial O_2 come down.
14	Door	To provide insulation over the gas tight seal.
15	Breather bag	To equalize pressure differentials due to temperature fluctuations and minor disturbances to the materials balance.
16	Orsat apparatus	To determine the atmosphere composition.

FIG. 3. . . .EQUIPMENT LOCATION FOR THE CONTROLLED ATMOSPHERE APPLE STORAGE

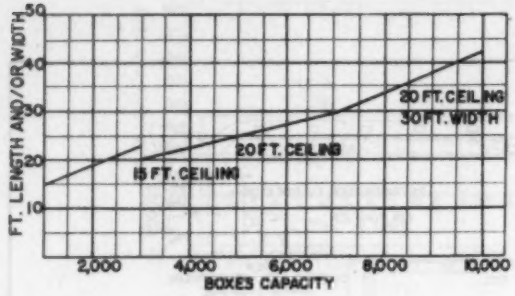


FIG. 4....RELATIONSHIP OF CAPACITY AND PHYSICAL DIMENSIONS

summarized in the table which accompanies Fig. 3. The identification numbers are keyed thereto.

CONTROLLED ATMOSPHERE STRUCTURES

Critical factors to be considered in a structure for controlled atmosphere are: Size, insulation, vapor barriers, gas seals and gross gas permeance tests prior to loading with apples. The relationship between capacity and length, width, and height is shown in Fig. 4. These dimensions are based on an allowance of 2.5 cu ft per box of apples and do not consider building or pallet modules.

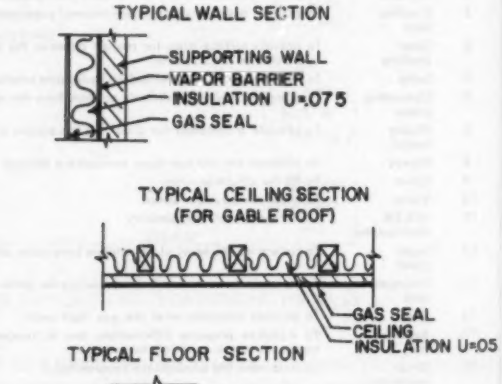


FIG. 5....TYPICAL SECTIONS FOR CONTROLLED ATMOSPHERE STORAGES

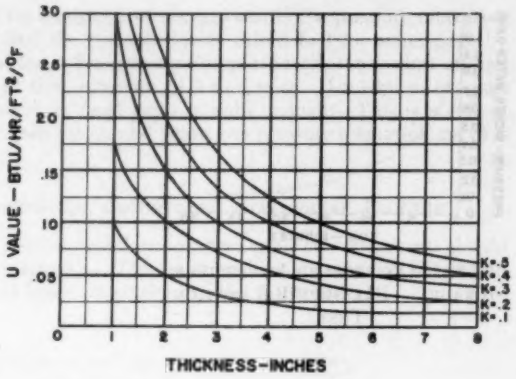


FIG. 6. . . .THICKNESS *vs* COEFFICIENT OF HEAT TRANSFER FOR INSULATION WITH CONDUCTIVITY FROM 0.1 TO 0.5 BTU PER (HR) (SQ FT) (F DEG) IN THICKNESS

Good practice in the use of insulation,[3] vapor barriers, and gas seals is illustrated in the typical floor, wall, and ceiling sections of Fig. 5. It should be noted that insulation requirements are specified in terms of U, the overall coefficient of heat transfer. A rapid means of comparing thicknesses of insulations of different thermal conductivities, K, to achieve equivalent U values is presented in Fig. 6.

Gross gas permeance tests after completion of the construction are the best assurance of the satisfactory performance of a controlled atmosphere storage. These gross permeance tests require a considerable amount of equipment in the way of air pumps and flow meters and their widespread use is unlikely. Gross permeance data can be used to compute a materials balance which will indicate whether or not O_2 leakage into the storage is at a rate in excess of what the apples require. If the leakage rate is in excess of the respiration requirement, then the 3 percent O_2 level cannot be maintained.[4] An example of two gross permeance tests appears in Fig. 7.

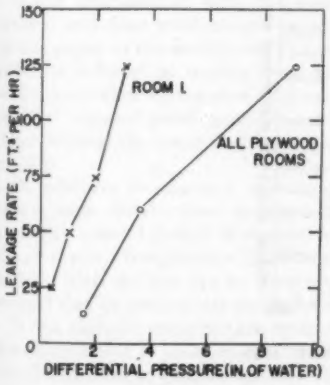


FIG. 7. . . .GRAPH SHOWING RELATION OF DIFFERENTIAL PRESSURE AND LEAKAGE RATE

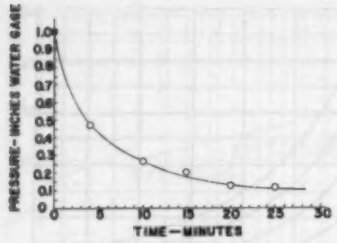


FIG. 8. . . .CONTROLLED ATMOSPHERE STORAGE PRESSURE-RELAXATION TEST

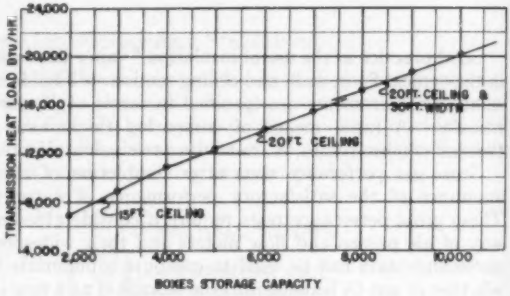


FIG. 9. . . .CAPACITY *vs* TRANSMISSION HEAT LOAD

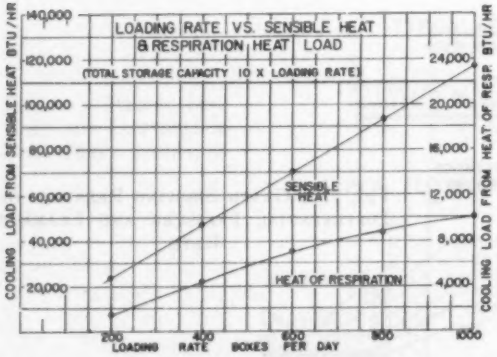


FIG. 10. . . .LOADING RATE *vs* SENSIBLE HEAT AND RESPIRATION HEAT LOAD

The commonly used test for adequacy of the gas seal is the pressure relaxation test.[5] The room is sealed and the pressure inside raised to 1-in. water gage by means of a small centrifugal fan. The pressure drop after the fan is shut off and the port closed is recorded at time intervals of 3 to 5 min. If 0.1-in. water gage remains after 25 min, the room in most cases is tight enough. This is a rule of thumb and has been widely used for several years. A pressure relaxation curve is illustrated in Fig. 8.

REFRIGERATION SYSTEMS FOR CONTROLLED ATMOSPHERE PROCESS

Refrigeration systems for the controlled atmosphere apple storage process should provide adequate cooling capacity. The relationships between refrigeration capacity and the various heat loads are shown in Figs. 9, 10, and 11.

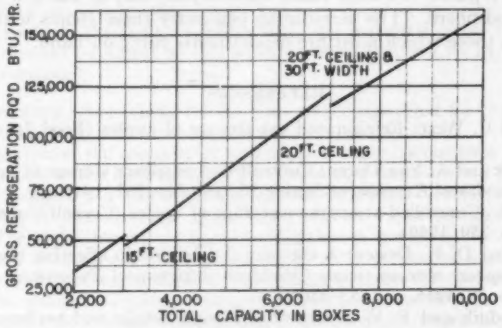


FIG. 11. . . . TOTAL CAPACITY *vs* GROSS REFRIGERATION REQUIRED

These relationships were developed on the basis of the specifications for wall, ceiling, and floor sections and geometric configuration of the structure presented in this paper in the section on "Controlled Atmosphere Structures". Sensible heat removal is based on cooling from 80 to 32 F in 24 hrs. It should be noted that while controlled atmosphere process conditions call for an operating temperature of 37 F, current good operation practice requires cooling during loading to 32 F. After sealing the controlled atmosphere room, the temperature is allowed to rise to 37 F.

In addition to adequate cooling capacity, refrigeration systems for controlled atmosphere should meet several other requirements. A reasonable amount of humidity control should be assured by provision of sufficient evaporator surface at the dew-point temperature (33 F for 85 percent relative humidity and 37 F dry-bulb). Mist nozzles can be used to correct certain small deficiencies in humidity control due to inadequate evaporator surface areas and temperatures.

While high air velocities are desirable during initial cooling, these high air velocities later result in unnecessary heat loads on the storage and in some cases excessive infiltration of outside air into the storage due to the aspirator effect of high

velocities across any pin hole leaks in the storage gas seal.[6] In some cases it has been necessary to install two speed blowers on evaporators or to operate evaporator blowers only part time in order to prevent excessive infiltration of outside air. Duct air distribution systems appear to be more dependable in this respect than do simple free air discharges on blowers.

To minimize the risk of off flavors or tainted fruit, it is justifiable to eliminate the use of any toxic refrigerant. This is a particular hazard considering that as a rule controlled atmosphere storages are completely sealed for at least 5 months and a small refrigerant leak in the evaporator could go unnoticed for some time.

Expansion valves should be mounted outside the controlled atmosphere storage for accessibility. Also refrigerant lines and electrical conduit should be grouped and fed into the storage at one central point to minimize the number of openings in the gas seal.

Evaporators and blower motors should be located near the access panel to facilitate emergency repairs. In some cases, these repairs can be made without loss of the storage atmosphere. The serviceman can enter these rooms with an oxygen mask similar to those which most fire departments carry on hand.

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[5] J. W. Zahradnik and F. W. Southwick: Design details and performance characteristics of a Douglas fir plywood CA apple storage (*Massachusetts Agricultural Experiment Station Bulletin* 505, 1958).

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DISCUSSION

J. R. SWANTON, JR., Cambridge, Mass.: What material and what material thickness were used for the breather bag? And also were any plastic paints used?

E. J. ROBERTSON, Chicago, Ill.: Were any bacteriological problems encountered while getting the process established?

LEE LIEPIN, New York, N. Y.: What temperature was found from the author's experience to be the best for apple storage and what were the allowable differentials? Also does the author have any recommendations to make for 2-stage refrigeration equipment to maintain these temperatures?

W. O. WALKER, Coral Gables, Fla.: What gas seal materials were employed?

LEON BUEHLER, JR., Chicago, Ill.: What about the skill of the operator? Is that any problem? Can the ordinary small storage afford the type of man that might be needed to run the plant? Is there anything critical about the type of package, wrap-

ping and perhaps the amount of air movement in relation to the controlled composition of the atmosphere? What I have in mind is whether a relatively tight package requires a different treatment, and what might be the effect of insufficient air movement.

S. J. STACHELEK, Rochester, N. Y.: By using some of the more modern insulating materials this discusser has erected rooms that people have called *too gas tight*. In some ways these rooms have brought about economies in operation and in other ways absolute gas tightness must have created new problems. Could the author evaluate that situation?

Secondly, in the building construction analyzed, the one with the truss gussets, how did the builder get a vapor barrier on the top side of the insulation?

R. L. SUMMERHAYS, Rochester, N. Y.: A new test room has been built at Ithaca, N. Y., using 8 combinations of different materials and procedures for obtaining a gas seal. The insulation on the walls was applied, using 4 different specifications. This room was recently completed and air tested.

One of the problems that has come up is the possibility of eliminating sodium hydroxide from the washing procedure. Some work has been done on aerating the wash water outside the room. Is the author familiar with any materials, other than sodium hydroxide, which could be added to this wash water to improve its capacity for transferring carbon dioxide?

J. G. MACORMACK, New York, N. Y.: Two subjects covered by the author are the basis of these questions.

One is the matter of the necessity of an adequate vapor barrier over the ceiling insulation construction in the installation which the author has shown. This installation, as he has pointed out, is a ventilated, gabled roof structure. There is a hazard in the recommendation of just a vented gabled roof with no vapor barrier over the ceiling insulation. The definition of adequate ventilation is questionable. Also, it would seem that these storages are being employed in more prolonged seasonal runs, with operation continuing for many more months than in the past with normal apple storage. This brings the ceiling insulation construction back to about the same hazardous position as a normal cold storage construction, if an adequate vapor barrier is not employed. Is this opinion agreeable with the author?

The second subject the author mentioned regards installations in Michigan, where no barrier is used in the floor. This proved satisfactory because metal liners sealed tighter and with much lower permeance than some laminate liners were employed in the walls and ceiling.

In New York State at Cornell University, the author will recall that controlled atmosphere facilities were installed incorporating aluminum sheet liners with tightly sealed joints, in walls and floors and with no gas barrier used in the floor insulation construction. From the point of view of economy, it would appear permissible to omit gas barriers in the floor, if the wall and ceiling liner are of sufficiently low permeance to provide the overall gross permeability of the structure.

A. G. WILSON, Ottawa, Ont., Canada: These questions are in connection with the permeability of the gas sealing material. Firstly, is resistance to air flow the main objective or is the permeability to CO_2 and O_2 of primary concern, or are both important?

Secondly, it appears that in many instances the application of the gas sealing material results in a better vapor barrier than that normally applied for condensation control. Is there concern about providing a vapor barrier on the inside of the storage that is better than the vapor barrier on the outside?

AUTHOR'S CLOSURE: The material in the breather bags as questioned by Mr. Swanton is a plasticized polyvinyl chloride. The breather bags are available commercially, and there is an information sheet obtainable from the University of Massachusetts that gives details of the dimensions and the source.

The various types of spray-on materials have been looked into only to the extent of getting what technical and economic information could be assembled.

What has been discouraging is the cost per square foot applied. The subject has not been studied too seriously because there are quite a few materials now that can be used successfully, and which fall into the price range of less than 7 or 8 cents per square foot, and which seem to do the job.

Mr. Robinson asked about bacteria. Every once in a while old apple boxes are used in the controlled atmosphere room. These old boxes are heavily loaded with molds and fungi, and if these boxes happen to be put in a very moist location there will be mold growth and some harm to the fruit, but this is the exception rather than the rule. As far as any bacteria are concerned, the author knows of no instance where they have been a factor.

As an engineer, I cannot authoritatively answer about the best temperatures and differentials. The best reply is to repeat what horticultural colleagues have stated in the course of cooperative work. For McIntosh the recommended temperature—and there is unity on this among the various pomologists—the current recommendation is for 37 F. The specification on differential for this temperature originated at the experiment station and is arbitrary because the larger the differential the larger the breather bag becomes; so the attempt is to keep the differentials to 1½ F deg. In most cases it can be done without great expense.

As regards the question concerning 2-stage refrigeration equipment, the author has had no experience where 2-stage systems were necessary. Most of the equipment worked with has been single-stage. Various compressor sizes have been so grouped that under ordinary loading and critical conditions one could throw on all of the compressors, and then after the initial sensible heat was removed, and as the winter months came on, one of the smaller compressors would carry the load.

A wide variety of materials has been employed. Historically galvanized iron was used by the pioneers. This whole process was stumbled on in England when apples were being shipped from Australia. It was found that, strangely enough, the apples kept better in the holds of the ships than they did in storage. This led to the conclusion that the gas-tight structure of the holds was responsible and thus sheet steel was being used as a gas seal. Since then, primarily because of economic considerations, the trend has been towards the newer and less expensive and less rigid materials.

A wide variety of films and laminates is being used. There is an aluminum mylar laminate which has been used successfully. Then there are in use various aluminum foil, kraft paper laminates. Also some water soluble emulsions have been used successfully. Also, high density plastic and medium density plastic overlaid plywood have been used. So also has plywood that has been painted with aluminum leaf paint.

Concerning packaging requirements, the author cannot cite any direct experimental data to support the statements, but can give some observations and what he has learned from work at other stations.

On the West Coast a considerable amount of work has been done with Delicious apples in a box which had a gas seal in it. The permeability of this gas seal was selected to be such that approximately the right materials balance could be maintained and approximately the right composition in the box. This reportedly did not work out very satisfactorily.

Gas analyses at several different locations inside of a given storage have been made by inserting the tubing in the storage when it was being loaded and running the tubing through gas-tight ports outside and then analyzing the atmosphere in the storage at several different locations. This was done in connection with some air circulation studies. The thought was that if one cut down on air circulation there might be stratification of the atmosphere. Very little difference in the atmosphere composition was found even with reduced air circulation.

Thermocouples had been placed in some of those apples. Some hot spots developed. So it seems as though the critical factor on air circulation is not atmospheric composition but the heat generated by the apples during their respiration process.

As regards Mr. Buehler's questions concerning the skill of the operator, the answer depends on the opportunities that the operator has for training. Most fruit farms are of such a nature that they must have at least one on-the-ball person around and he can be taught to make the Orsat analysis and to run a controlled atmosphere room satisfactorily. It is a very risky matter. With 30,000 boxes of apples in a storage valued at $5.00 per box, the rather large investment should not be entrusted to someone who is not reliable and dependable. Care needs to be taken as to who is going to run the room. This simply emphasizes the need for reliable automation.

No attempt was made to put a vapor barrier on top of the insulation. A gable roof construction was used and the gable was well vented, the idea being that this is a cycling process. Whenever one has this problem of moisture condensation in insulation and in the attic, which is quite accessible and has a good opportunity for air exchange, if there were any condensation it would dry up in the off-cycle.

As mentioned, samples of insulation from the storage have been taken and they have not varied very much at all. Samples are taken from the ceiling as well as from the wall.

In answer to the question of Mr. Stachelek, offhand it would appear that one could not get a room too tight. This thought was not from economic considerations but rather from consideration of the materials balance because one has to control this process and if he can control it through a valve which he opens and closes, or through a switch which he turns off and on, this is a much more reliable control than one that depends on the permeability of the walls.

This does not eliminate from consideration the possibility of a completely automatic room that is so designed that the materials balance can be maintained simply by permeabilities. This is something which is yet to come—a fully automatic room—and this is one approach to it. That is, once steady state conditions are reached, breather bags of such size and permeability would be provided that take out the right amount of CO_2 and add the right amount of oxygen. This is theoretically possible, but is not practically possible at this time. But it suggests a means of automating this process.

What is being done at the present time on an experimental basis both in New York State and in Massachusetts at the experiment stations is work that perhaps eventually will eliminate the use of sodium hydroxide as an absorbing liquor, except perhaps on the initial come-down when the rate of CO_2 production is very high, and substituting water for the sodium hydroxide.

This is an absorption-desorption phenomenon and the design of the scrubber needs to be different from the one which uses sodium hydroxide. Sodium hydroxide and carbon dioxide react readily, so all that is needed is an opportunity for that reaction to take place. It is the kinetics of the reaction that determines primarily the characteristics of the scrubber, and the biggest variable found in CO_2 scrubbers that use sodium hydroxide has been the sump capacity.

For a larger room it is only necessary to add more chemical. It is not necessary to adjust the air flow rate or the liquor flow rate. When water is used, however, this flow rate of the equipment in relation to the size and capacity of the storage becomes much more important. The flexibility in scrubbers present when sodium hydroxide is used is no longer available if water is used as an absorbing liquor. It appears now that both the inside type scrubber, which New York State has been using, and the outside type scrubber, which the Massachusetts experiment station has recommended, can both be used with water after the initial come-down.

Whenever something is added to water to make it take up more CO_2, a chemical question arises, *viz, What is the best chemical that can be put into water to make it pick up CO_2 faster?* The cheapest and the one that has been used in the past was sodium hydroxide.

This practice is hard on the metals. There is a corrosion problem, particularly on the inside type scrubbers. The Massachusetts station has not recommended the inside type scrubber because of the hazard to the evaporator. In using the outside type scrubber, an all cast-iron pump can be used and an all iron blower. So if there is any vapor carry-over there is a certain amount of corrosion resistance. Many of the scrubbers

used in New England have been made of black iron of a rather light gauge and they have given 4 or 5 years service without any great corrosion problems. Caustic soda is not very corrosive to black iron.

With regard to the question of Mr. Macormack, this is simply a matter of the gross permeability of the structure. In other words, a certain number of leaks can be tolerated so long as the oxygen requirement of the apples is not exceeded. It is very difficult to answer the question in general terms because of this particular factor. Even the size of the storage enters into it.

Concerning use of a test for leakage rate with the pressure falling from 1.0 in. to 0.1 in. in 25 minutes, offhand, it could be well used as the yardstick. There is one consideration that sometimes makes the evaluation of the behavior of these storages complex and that is the air movement. There is no intent to belittle that point at all, but it is very important. One who has ever tried to run a pressure relaxation test on a room and then put the fans on at the same time will surely have found that the room really leaks much more whenever there is movement inside the room.

One other factor was not mentioned about the pressure relaxation test. One should test this room before turning the refrigeration on. The room should be essentially at the same temperatures as the ambient. Otherwise relatively warm outside air will be introduced into a room at 32 F, and the warm air will shrink to an extent that no pressure will build up for a good while. It is very difficult to test a room once it has been refrigerated. Even the walls, if they are cold, will cool down the air pumped in so rapidly that it is difficult to evaluate this test. So the room should be essentially at ambient.

To answer the first question of Mr. Wilson, there is interest in both the resistance to air flow and the permeability. The permeability of the material itself is important but it is not the most important. The gross permeability of the room is the most important because, as mentioned before, one can have a very fine material and literally ruin it because he doesn't take care of the joints when he puts it up. Thus the interest is in both.

The question on vapor barriers needs to be recognized. The primary consideration when controlled atmosphere storage is the subject has been the maintenance of a relatively impermeable gas seal on the inside of the wall and in most cases the barrier on the outside of the wall is not necessarily designed for gas tightness against the atmosphere. It is designed as the vapor barrier for the insulation and the atmosphere is contained in the storage by the gas seal which is on the inside of the storage, the inside envelope, and also the vapor barrier on that particular side.

1688

No. 1688

PROPOSED ASHRAE STANDARDS FOR THE MEASUREMENT OF SOUND FROM EQUIPMENT

By C. M. Ashley*, Syracuse, N. Y.

THE PURPOSE of comfort air conditioning is to provide a more favorable environment for people. More specifically, it deals with the conditioning of the air surrounding people. The definitions of air conditioning include the control of the temperature, humidity, cleanliness, mass motion and quality (with respect to contaminants).

But, they omit any mention of one important characteristic of the air: the vibratory motion known as sound. Objectionable sound can come from a wide variety of sources, including traffic, typing, talking, mechanical equipment and even from air conditioning and refrigeration. Contributing to the sound can be such factors as poor building construction, improper sound treatment of offices, flimsy partitions, cracks under doors and many other factors which permit undesired sounds to reach the ears of building occupants.

Absence of sound can be just as objectionable as too much or the wrong kind of sound. Sounds of good quality can be used to mask out objectionable noise and can protect private conversations and prevent intrusion of traffic and typing noises. Thus, it is beginning to be learned that what is needed is not sound removal, but sound control. In fact, *sound conditioning* may be considered as the new tool for improved human comfort.

Air conditioning which includes elements providing for heating, ventilating and cooling can and should be an important part of sound conditioning. But, frequently the air conditioning is charged with the overall responsibility for the sound conditioning. Other factors including building construction, other mechanical equipment and the installation of the air-conditioning equipment must bear a major share of the responsibility. An important reason for knowing the sound output of the air-conditioning equipment itself is so that a proper division of responsibility can be made between sound which is made by the air-conditioning equipment and sound coming from other sources.

The Sound Standards Program

The sound standards program of the Society is the result of a growing recognition of the fact that one cannot hope to get proper sound control with respect to air

* Chief Staff Engineer, Carrier Corporation. Member of ASHRAE.
Presented at the Annual Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Lake Placid, N. Y., June 1959.

conditioning and refrigeration without having adequate means of objective measurement any more than he could get adequate temperature and humidity control without being able to measure the cooling and heating power required to effect them. The fact that there is not universally adequate control over air-conditioning sound is amply attested by the results of user surveys such as those of duPont[1] and Architectural Forum[2] and by our own personal experiences with noisy equipment.

As an important step in overcoming this situation, ASHAE in 1955 activated a sound study program in its Research Laboratory. The initial project was to determine the techniques of measurement of sound power output of fans.[3] The program is being continued at the present time with a study of sound generation and attenuation in duct systems. In 1957, ASRE took the leadership in setting up a standards committee to prepare a standard for sound testing of equipment and invited ASHAE to participate in the standard on a joint basis. A task committee was appointed in July, 1957, and has prepared several drafts of a primary test standard.

In accordance with recognized practice, the Society Standards Committee writes a standard test procedure, but does not ordinarily set up equipment ratings or other recommended practice. In order to make the standards activity effective, additional steps are required of trade associations or other organizations. Among the steps which are pertinent to a sound standard are:

1. A recommended basis of rating for different types of equipment.

2. Interpretation of the ratings in terms which are meaningful to the user of the equipment.

3. Recommended sound criteria for different space uses.

In undertaking its assignment, the ASHRAE committee was fully conscious that there was a need for much more widely diffused knowledge and understanding of the technical aspects of sound and that there must be a step-by-step development of the procedures rather than attempting to reach the ultimate immediately.

However, it was also recognized that if the standard or standards were to be of value, they must meet at least two criteria. First, the performance of equipment should be measurable in terms which would be significant regardless of the environment. Second, the results must be capable of reasonable correlation with the subjective impressions of observers in spaces where the equipment is to be used.

FEATURES OF THE STANDARD

Sound power. In order to satisfy the first of these requirements, it was agreed that the measurement of sound must be stated in terms of the sound-power output of the equipment rather than the sound-pressure level[4*] which has been customarily used in the past and which is the basis of most sound measurements in rooms. This is necessary because sound-pressure level is a function not only of the sound output of the equipment, but also of the sound characteristics of the space in which the equipment is located. Using sound-pressure level as a basis for rating equipment sound would be like using room temperature as the basis of rating the capacity of a

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[1] Superscript numerals refer to References at the end of the paper.

* For definitions of acoustical terminology as well as much valuable background information, see Chapter 23—Sound Control, in the HEATING, VENTILATING AIR CONDITIONING GUIDE, 1959.

cooling unit. Neither one deals with the performance of the equipment itself, but only with the results in a specific environment.

Because the total variation in the sound-power output of different types of equipment is great and because the ear responds to sound on an approximately logarithmic rather than a linear scale, the power output of the equipment is expressed in logarithmic terms rather than in arithmetical terms as sound-power level.* If the sound power output of the equipment and the acoustical characteristics of the room are known, then it is possible to make a reasonably accurate prediction of the sound-pressure level to be expected as a result of the equipment sound just as it is possible to make a reasonably accurate prediction as to the room temperature which will be achieved with a given cooling unit under given ambient conditions.

The sound-pressure level of a room is usually in the range of 12 to 20 db below the sound-power level of equipment radiating sound directly to the room.

Sound quality discrimination. The reaction of a person to the equipment sound is the result not only of the sound-pressure level, but also of the frequency, time and space characteristics of the sound. In general, the characteristics of a sound which the ear recognizes as good are:

1. *Broad band,* that is, having a continuous and fairly uniform frequency spectrum without noticeable single frequency or pure tone components.

2. Without special character or pitch to make the listener conscious of the sound. This effect is produced by broad band noise having decreasing sound-pressure level per octave for successively higher frequency bands.

3. Continuous with respect to time rather than intermittent or varying in level.

4. Well diffused rather than being directional or concentrated close to the observer.

Fortunately, most sounds from aerodynamic sources, such as fans, tend to conform to the description just given for a good sound. Thus, the major problem with such equipment is to be able to maintain a proper level and to eliminate sources of poor quality noise generation such as whistles due to air leaks, blade cut-off noise, and magnetic motor noise.

However, the sound from refrigeration equipment tends to have single frequency components due to such causes as motor hum, gas pulsations, and mechanically excited vibrations. Thus, the noise from the mechanical parts of the equipment are likely to be far less desirable in quality than is the air noise.

Frequency bands. In order to provide a measure of quality, the committee is recommending the determination of the sound-power level on an octave-frequency-band basis. When these values are converted to room sound-pressure level, the effect of the sound in the room may be interpreted in terms of various criteria such as the Noise Criteria,[5] the Neighborhood Annoyance Index,[6] Speech Communication[7] or Loudness[8] since all of these criteria are based upon octave-band room sound-pressure levels.

It is recognized that the presentation of the data in octave bands is not necessarily a complete solution to the problem because the octave band readings may not fully reflect the annoying quality of discrete frequencies, beats or time or space variable sounds. Nevertheless, it is believed that it is a good beginning and that it is particularly appropriate in view of the prevailing characteristics of sound from air-conditioning equipment.

Quite possibly at some later date after experience has been gained with the use of the proposed standard and after analytical test equipment is more generally available it may be desirable to change the requirement to a smaller band width, such as one-third octave.[9] Also, it may be found advisable to add a requirement for a narrow band analysis to show up the presence of discrete frequencies. However, it is believed by your committee that neither of these steps is appropriate to the present situation.

Test room requirements. The committee next turned its attention to the question of the type of environment in which the test should be run. From the practical point of view, there are a number of factors which need to be considered. *First*, the cost of the test facilities should be kept as low as practicable consistent with good accuracy in measuring the sound. *Second*, the method of testing chosen should be simple enough to be usable by a person having relatively little background in sound testing and also should be relatively economical of test time. *Third*, the test space should be as flexible as possible for the testing of different types and sizes of equipment and should permit as far as practicable the use of existing facilities. *Fourth* and most important, the test facility together with the test technique chosen should be such that the results obtained provide the true power level and give re-producible results between different laboratories.

Anechoic room. Use of an anechoic room was considered. In such a room it is possible by taking a series of readings on an envelope around the equipment to obtain the sound-power level directly from the readings of sound-pressure level by taking into account the area of envelope on which the measurements are made. However, in an anechoic room one is faced with the same sort of problem in measuring sound as when measuring air quantity with a Pitot tube in a duct: to have good accuracy, a number of readings must be taken. In the case of the envelope measurement, it is probable that at least 12 stations would be required. Thus, unless automatic traversing equipment is available, the measurement becomes laborious and complex. In addition, the cost of an anechoic facility is quite substantial and it is not readily adaptable for or from other uses.

Semi-reverberant room. The committee next considered a *semi-reverberant* room which has a fair amount of sound absorption, possibly comparable to that in a normally occupied room and where both direct and reflected sound is received by the microphone.[10] Such a facility has an advantage in comparison with an anechoic room of lower first cost and greater utility for other purposes, but it will still require the use of a multiplicity of stations to provide the desired information. It is recognized that either an anechoic room or a semi-reverberant room is necessary if the directivity characteristics of the sound are to be measured. However, the committee felt that at least initially it was not desirable to include this additional requirement.

Reverberant room. The committee then considered the use of a reverberant test room in which it is possible for the sound to be averaged by multiple reflections from the relatively hard surfaces of the room. In such a room it should be possible to obtain the required data by means of a single station rather than the multiple stations of the traverse in the anechoic and semi-reverberant rooms. The reverberant room is essentially a room of suitable size and shape constructed of relatively hard nonabsorbent surfaces. However, for most air-conditioning and refrigerating equipment there will also be a need to provide unusually good isolation from outside noises so that they will not interfere with the measurement of the

sound from the equipment. This usually involves the use of a room of basic masonry construction resiliently mounted and possibly enclosed in an outer room.

There are a number of problems in connection with the testing of equipment in a reverberant room. Unfortunately, the sound is not reflected and diffused uniformly all over the room.[11] Particularly at low frequencies, the room tends to act in the same manner as an organ pipe, the sound being reflected back and forth between opposite surfaces. Since the room is three dimensional, these reflections may occur on a one dimensional or two dimensional or three dimensional basis and the resulting modal distribution patterns become quite complex.

At low frequencies there is an appreciable frequency separation between successive room modes and in order to space these as evenly as possible, it is important that the proportions of the room be kept within prescribed limits. In order to increase the number of modes at low frequencies, it is desirable to have the room as large as practicable. Even at higher frequencies, a single frequency sound produces a standing wave pattern in the room. Thus, it is desirable to have a swinging microphone which moves back and forth so as to be able to average the sound over an appreciable path length.

Microphone location. The microphone should be a reasonable distance away from the room surfaces because there is an increase in pressure level adjacent to the room surfaces and a further increase next to the edges between two surfaces and a still further increase in the corners. In addition, the microphone must be located far enough away from the equipment so that it will be in the *reverberant field*, that is: so that the amount of sound which is radiated directly from the equipment to the microphone will be negligibly small in relation to the amount received by the microphone from the multiple room reflections.

It is fortunate that the sound being measured is generally broad band in character since the effect is to provide an averaging of the sound which largely submerges the unusual characteristics of the room. However, it must be recognized that where single frequency components are prominent in the sound spectrum of the equipment, the measurement in a reverberant room is less than ideal. This is particularly true of single frequency components in the lower frequency region below 150 cps. A highly reverberant room may react with the equipment either to increase or diminish the sound power of the single frequency component. The magnitude of these effects is decreased as the amount of sound absorption in the room is increased. Thus, it is desirable to use as much sound absorption in the room as practicable and still retain the reverberant character of the room.

In order to satisfy the requirements as listed, a minimum room volume of at least 6,000 cu ft is proposed together with room dimensions of approximately 25 ft long, 20 ft wide and 12 ft high. Larger rooms are preferred and where the equipment size is large, may be necessary. A suggested series of dimensions are given in the standard and are shown here in Table 1. The table also recommends optimum sound absorption characteristics for the room.

Diffusion of sound in reverberant room. While it is accepted that the basic reverberant room recommended in this discussion is adequate for most of the sounds which will be measured, it is also recognized that in looking toward the future some additional means of diffusion must be provided.[12] One method of accomplishing this result is to increase greatly the size of the room while retaining good proportions and sound absorption characteristics.

Two other approaches are being considered which are applicable to the smaller sized rooms. The first of these, which has been used in a number of acoustical

laboratories, is to provide a large revolving reflecting vane which continually changes the effective room geometry through a substantial range. While it is probable that satisfactory results can be achieved by this means, it requires the use of a vane of dimensions not much smaller than the room dimensions. Thus, there is little room for the apparatus to be tested particularly so if it is bulky. In addition, there are considerable mechanical and acoustical complications in providing a large rotating vane which will also not generate measurable noise.

TABLE 1. . . .TEST ROOM CHARACTERISTICS

VOLUME, CU FT	RANGE OF DIMENSIONS, FEET			MINIMUM DISTANCE MICROPHONE TO ACOUSTIC CENTER[d] OF SOURCE, FT	MAXIMUM EQUIPMENT PERIMETER, FT	ROOM ABSORPTION CONSTANT R, SQ FT
	LENGTH	WIDTH	HEIGHT			
6000[a]	28–25	18–20	12[c]	13	20	150–210[e]
7500	25	20	15[b]	14	20	175–250[e]
	32–28	21–24	15			
10000	27½	22	16½[b]	15	25	200–300[e]
	25	21½	18			
	42–39	32–34	12[c]			
16000	36	28	16	20	33	300–450[e]
	32	26	19[b]			
	51–47	41–45	12[c]			
25000	46–43½	34–36	16	25	50	400–600[e]
	45–38	28–33	20			

[a] Minimum volume.
[b] Most desirable proportions.
[c] Minimum height.
[d] Acoustic center of source is the projection of the physical center line of the equipment to the nearest room surface or the intersection of the two nearest room surfaces when close to both.
[e] Room absorption constant R corresponds to a range of sound absorption coefficients of 0.07 to 0.10. The sound absorption coefficient and room constant should be a maximum in the first and second octave bands and should be less than the range indicated in the higher bands. This absorption characteristic can be obtained by the use of hard, relatively light panels well spaced away from the walls. The room absorption constant can be determined experimentally with a calibrated standard sound source and sound level meter according to Section 2.4 and Equation 4

$$L_w = L_p + 10 \log_{10} (R/4) - 0.5 \text{ db} \quad (4)$$

A second approach to the problem is to provide bumps or panels to increase the number of effective reflecting surfaces in the room. If these panels are to be effective for the low frequency sound, their dimensions must be a substantial part of a wave length, probably of the order of at least 8 ft or 10 ft. In addition, there is some uncertainty as to how far out from the walls they must be placed in order to be effective. There is a need for further research on means of diffusion and it is hoped that before the standard is ready to be issued a better evaluation will have been possible.

Accuracy of readings. If equipment ratings are to have any real significance, they must have a reasonably high degree of absolute accuracy or at the very least must be capable of being checked in a series of laboratories. Since the differences in competitive equipment are likely to be of the order of 5 db or less and since a 5 db variation in sound-pressure level is readily observed by a listener, this large a plus or minus variation would not be acceptable. A probable variation of ±2 or 3 db would still be undesirable, but possibly acceptable.

The need for accuracy poses a serious technical problem since there are a number of possible sources of error, some of which can be of substantial magnitude. These include deviations in the sound-level meter and octave-band analyzer used for the sound measurement, the variations of sound-pressure level in the reverberant room, effects due to the frequency characteristics of the equipment, observational errors and a number of others. In the past it has been the practice to determine the sound absorptive characteristics of the measuring room and then determine the sound-power level indirectly from these characteristics and a measurement of the sound-pressure level. Such an indirect method involves all of the major sources of error listed.

Standard sound source. In order to improve accuracy and eliminate as many sources of error as possible, the committee has adopted a new method, which involves the determination of the sound-power level directly by comparison with a calibrated standard sound source.[1] This result is accomplished by measuring the octave-band sound-pressure levels of the calibrated source and of the equipment in the test room. The sound-power level of the equipment is then determined directly as the difference of the equipment and standard-source pressure-levels plus the standard-source power-level. For instance, if the sound-pressure level reading of the standard source is 70 db and the reading of the equipment is 50 db, then if the source power level is 85 db, the equipment power level will be 65 db.

While a variety of sound sources are available for such comparison, the source which is favored by the committee consists of a small, direct-motor-driven fan wheel with restricted inlet. This fan has the advantage of providing a broad band source of sound of nearly uniform level for all 8 frequency bands. At the same time it is small and relatively portable and can be operated on readily available 60 cycle a-c current.

One of the most attractive features of such a source is its high degree of stability. The output should remain virtually constant over a long period of time, subject only to mechanical damage or the accumulation of dirt, both of which are readily observable and can be corrected. Minor corrections of the sound-power output for barometric pressure, temperature and speed can be provided if desired. The speed can readily be measured by means of a stroboscopic disk and neon light. The sound-power output of the standard sound source may vary slightly between a location in a corner or in the middle of the room. However, by calibrating in the position of use, this possible source of error can be eliminated.

PRIMARY STANDARD

It is intended that the primary standard be developed for equipment which will radiate sound to the air in a room or to the atmosphere outside. This category includes a wide variety of equipment ranging from room air conditioners and room fan coil units through self-contained air conditioners, ventilating fans, unit heaters, cooling towers and refrigerating equipment to large refrigerating machines of the reciprocating, centrifugal and absorption types. It is accepted that a single standard can be used for all of this equipment. The primary differences will be in the size of the test space required and the degree of isolation from outside noises. Since the basic standard can provide for such differences, it should be possible to cover the complete range with one standard.

A sub-standard of this standard can provide for the special problems involved in connection with room air conditioners, attic units and the like in which a portion of

the equipment radiates to a room and another portion radiates to the outside. The principal additional requirements for this sub-standard are a description of the conditions of mounting and of the treatment of the space surrounding the other part of the equipment. The sound-power level will be determined first for one side and then the unit will be reversed and the sound-power level determined for the other side.

Fan sound standard. A second standard is needed to cover the sound-power level transmitted by fans through ducts. Two general approaches to this problem have been considered. The first is to measure the sound-power flux through the ducts by measurements taken in the ducts. The second is to measure the sound power flux from the ducts by measurements taken in a room in which the duct terminates. There are technical problems involved in obtaining a correct result with each of these methods. However, the present thinking of the committee tends to favor the use of the second method because it is more consistent with the procedure for measurement of radiated sound and because there appear to be fewer problems in connection with this method.

Perhaps the most important of the problems which must be solved is to prevent the low frequency sound from being reflected back through the fan from the duct termination. In order to accomplish this result, it is proposed to provide the termination with a large horn (expanding section) which will act as a transformer to match the impedance of the duct to the impedance of the room, thus minimizing the amount of reflection.

Other problems have to do with a suitable termination of the duct on the other end of the fan, arrangements for control and measurement of the air quantity and pressure and the measurement of the sound radiated from the fan casing itself. These problems are being considered by the committee and it is hoped to have a tentative standard in draft form in the near future.

Terminal (outlet) sound standard. A third category of equipment is that of duct terminals such as outlets, grilles, induction units and the like. These terminals serve both to attenuate sound and also to generate it due to the flow of air through the terminal. Thus, to describe completely the sound characteristics of the terminal, it is necessary to provide a double set of information.

The amount of sound generated by the terminal can be measured readily by supplying the terminal with a quiet source of air and measuring the sound radiated using the same techniques as those described for radiation from equipment.

The same set-up can also be used for measuring the amount of sound transmitted from the ductwork through the terminal by cutting off the air supply and feeding sound through the duct connected to the terminal by means of a loud speaker. The sound flux entering the terminal from the speaker can be measured by removing the terminal and substituting for it a horn to act as a transformer so that the impedance of the duct will be coupled properly to the impedance of the room. Using this technique all of the readings will be taken in the same manner and will directly compare the sound power from the equipment with the sound power of the standard source.

FUTURE STEPS

What are the future steps which must be taken to put this series of standards into practice? The next step is a wider circulation of the primary standard in draft form to obtain the comments and criticisms of persons and organizations who may

be interested in its use. The primary purpose of this paper is to provide further explanation and background to go with the standard when it is circulated and to explain the background of the standard to members so that they will be in a better position to understand it, to make suggestions and to use it when it is issued. There may be a place for further discussion at the national or local meetings of the background of the standard and of many of the problems which are involved in connection with the sound testing of equipment. An important part of the proving out of the standard will be the testing of the same piece of equipment in a series of laboratories using the practices recommended in the standard.

After the standards committee has received comments on the draft of the standard, it will be revised again in light of the comments received and will then be ready for formal submission to the Standards Committee of the Society. Following a review by the Standards Committee, the proposed standard will then be published in the ASHRAE JOURNAL or otherwise presented to the entire membership for comment. After having received and incorporated the comments received from this step, the standard will then be ready for formal approval. This process will probably take between 1 and 2 years.

Cooperation with other groups. In the meantime the committee is working actively on a cooperative basis with other groups. Among these are the committee of the American Standards Association which has written a proposed generic equipment sound-testing standard[14] which has been used in large part as a starting point for the present standard by the Society committee; a committee of the American Institute of Electrical Engineers which is writing a standard for the sound testing of small, medium and large sized motors; and a committee of NEMA which has been designated to write a sound standard for room air conditioners. The committee is also cooperating with committees which have been set up by trade associations, ARI and AMCA which are working in parallel to prepare practice procedures for the sound rating of air-conditioning, heating and refrigerating equipment and the interpretation of the rating in terms of acceptable levels for different applications.

By the time the test standard is completed and approved, it is to be hoped that the work of the trade associations will be advanced to the point where a timetable can be set up for putting the sound testing and rating into practice. It is also hoped that arrangements can be made with an independent testing laboratory to provide sound testing service to those manufacturers who do not wish to provide their own facilities.

If the steps being taken obtain general approval, it should then be possible to provide sound ratings of air-conditioning and refrigerating equipment. Thus, it should be possible for the first time to provide a clear division of responsibility: to the manufacturers for the sound output of the equipment which they manufacture and to the consulting engineer, architect and owner for other sources of noise and for the proper installation of the air-conditioning equipment.

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DISCUSSION

G. N. SAWYER†, Tyler, Texas: This is an opportunity to comment about specifying the noise of equipment by total sound power per octave band. This is a very useful concept. It is independent of the environment into which the equipment is put. Knowing the sound power one can calculate what the sound level will be in different surroundings. It is also useful for calculating sound transmitted from equipment through duct work. Those who have been in sound work for some time are convinced that this is the proper way to specify equipment noise.

How does the author feel about specifying the sound in one-third octave bands?

J. B. CHADDOCK, West Lafayette, Ind.: In looking at Table 1 for the first time some engineers feel that the room volumes specified are quite large, and at first encounter with reverberant rooms this is a natural reaction. The committee on Methods of Equipment Sound Testing, of which Mr. Ashley is chairman, spent many hours discussing this room volume problem. The reasons for the room sizes listed in Table 1 has been pointed out by the author, but perhaps not fully emphasized. He has mentioned that the establishment of modes of sound vibration in the reverberant room present a problem of accurate measurement. This is particularly true in the low frequency bands of long wave length and that is the reason for requiring a relatively large room volume. The choice of 6000 cu ft as the minimum room volume was for the purpose of having at least 10 modes in the lowest octave band. With this number of modes only a few measurements of sound-pressure level are necessary in order to obtain accurate measurements of the sound output of the equipment under test.

J. P. LAUGHLIN*, Evansville, Ind.: This might be more of a comment than a question, but the 6,000 cu ft minimum volume causes the realization that many existing sound rooms in the industry are much smaller than this in size. Also, there is usually a premium on factory space and that in high noise environments. One might well question

† General Electric Co.
* Whirlpool Corp.

whether the industry will support this requirement for a test facility particularly if there is any other way to accomplish the same purpose.

Is there any possibility of converting and utilizing existing rooms, perhaps by conversion factors, or by comparison to a true reverberant room?

E. H. JENSON*, Staunton, Va.: We have a room which is nearly the minimum size described and have run into the problems mentioned by the author. This experience confirms the point that the minimum may be too small. The ceiling is substantially lower than the author's and that may make a difference. Since it is an 8-ft ceiling, it would appear that the minimum should be higher than it is in the table.

C. R. HIERS, New York, N. Y.: What about the size of the equipment to be tested in the room? For example, a fan of 2 in. in diameter might be tested and possibly in a week a fan 22 feet in diameter using 2,000 hp might need to be accommodated. Should some parameters be laid down covering the relation of room size and size of test specimen?

A MEMBER: Could the author elaborate a little more on his comment concerning the shortcomings of mixing vanes as a means of eliminating room geometry?

AUTHOR'S CLOSURE: Mr. Sawyer has raised an interesting point. Personally, the author would be very much in favor of specifying sound power in one-third octave bands, but thinks that at the present time it would be difficult to obtain general industry acceptance of this method of reading and reporting sound information.

In specifying transformer noise, NEMA decided to use a weighing network instead of a frequency-band method on the basis that the weighing network gave a single number which was readily understandable and very serviceable. However, it must be remembered that transformer noise is composed almost entirely of a series of single frequencies comprised of a fundamental of 120 cycles and the harmonics of this fundamental. Thus, the noise from all transformers is rather similar in character. This is not true in the case of air-conditioning equipment in which there is a great difference in the character of noise from different types of equipment.

Dr. Chaddock has emphasized a very important point which is likely to be somewhat controversial because a room of the recommended minimum volume of 6,000 cu ft is fairly expensive and because some companies already may have sound rooms which do not conform to this recommendation. The committee believes that it is important to have a large test facility to get accurate readings. The proposed room is the minimum size which will provide 10 modes in the lowest octave band up to 75 cps and a considerably larger number in the next octave band.

The response of the room is sharply peaked at each mode. With a highly reverberant room, for instance, the frequency width of the modal response, 3 db down from the peak on either side, is only $\frac{1}{8}$ of a cycle per second. By increasing the amount of room absorption somewhat, the width of the response band can be increased to approximately $\frac{3}{4}$ of a cycle per second. Thus, there arises the problem of attempting to spread 10 $\frac{3}{4}$-cycle band widths over a total band width of 40 cycles. Obviously, there are going to be deep gaps between modal responses. The room specified is not as good as the committee would like it to be, but with improvement in the amount of diffusion, the committee believes it will be a workable compromise.

Mr. Laughlin asked about the use of smaller rooms. Unfortunately, with smaller rooms the sound-pressure level may be greatly increased or greatly decreased depending upon the particular characteristics of the room, *i.e.* the interrelationship of the exciting frequencies and the room modes and the position of the sound source with respect to the nodes and loops of the standing wave. For instance, in one room in which sound work was carried on, the room response happened to peak at 120 cps and a terrific standing wave resulted from the magnetic hum of certain types of equipment. On the other hand, the reverse can also happen. If the 120 cycles had been in a valley between two room

* Westinghouse Electric Corp.

modes, the amount of sound measured would have been far less, rather than far more, than would have occurred under free field conditions. Unfortunately, there is no way of correcting for this difficulty. It has to be accepted as an inaccuracy of the method.

From Mr. Jenson's comments, it is gratifying to see that there is also support for a larger room volume. The comment is undoubtedly right from a technical point of view. The author's own feeling is that approximately 10,000 cu ft is as small as the room should be to provide good results technically. But, it is also necessary to recognize that some compromise should be made in the interest of practicability and the feeling of the committee was that the 6,000 cu ft room represents a not-too-bad compromise. Where the sound to be measured is broad band in character, the 6,000 cu ft room will give very good results. However, for single frequency noise measurement, the 6,000 cu ft room represents a distinct compromise. There is need for further research to improve the diffusion of sound over the whole frequency spectrum. It seems quite certain that substantial improvement can be accomplished. It is not the thinking of the committee that the room will be a simple rectangular box, but rather that it will eventually include additional bumps and other features which will tend to improve the diffusion.

One method of improving diffusion which is used in a number of laboratories is a large revolving vane. This method sweeps the room modal response characteristics through the whole range of different frequencies, thus tending to even out the frequency response of the room and provide better diffusion of the sound. However, from the practical point of view, the revolving vanes present many serious problems and they also do not produce complete diffusion at low frequencies.

Another approach to the problem is to place diffusing elements around the wall. There has been a recent paper reporting a study made at a higher frequency level which gives quite a little encouragement as to what can be done to improve the diffusion by this method. The committee is very hopeful that further research can be done to produce a completely satisfactory room and some studies looking toward this end are now in progress.

Mr. Hiers suggested the subject of relating the room size to the size of the test specimen. Approximate values of the limiting sizes of equipment are shown in Table 1. The lower limiting size of the room is not limited by the size of the equipment, but by other considerations. On the other hand, it is obvious that a room of minimum size will probably not be able to take care of extremely large equipment.

Generally, it can be said that the perimeter of the equipment should be kept no greater than the largest dimension of the room. But, there is considerable latitude in the size of the equipment which can be used in a given room.

One of the things which is hoped to do in connection with the further evaluation of the proposed standard is to run a round robin test on equipment. As this project goes forward, much will be learned about the details of the method and the limits that may have to be applied to it. However, the information now at hand encourages the belief that the method as set forth will provide a realistic standard and that needed modifications to it will be relatively minor.

There was a request for further comment about the use of revolving vanes. There are some serious mechanical problems in connection with these vanes principally having to do with the supporting and driving of the vanes so that the mechanism does not make noise. One of the principal objections to the vanes is that they are so large that they will tend to interfere with equipment which is to be tested. In most of the laboratories which are now using them, they are used for testing of sound absorbing materials on the walls or for the sound transmission through wall panels so that this space problem does not exist. Entirely apart from the mechanical problems, however, the revolving vanes leave considerable to be desired for the measurement of equipment noise. For instance, where a beat exists between two closely spaced frequencies, it is doubtful if the vane will provide a proper picture of the characteristics of the noise.

1689

No. 1689

EXPERIMENTAL STUDY OF GRILLE NOISE CHARACTERISTICS†

By B. H. MARVET*, KNOXVILLE, TENN.

TO PREDICT characteristic patterns inherent in grille noise, three test grilles shown in Fig. 1 were compared under identical experimental conditions by relating octave-band spectra, loudness spectra in sones, and total loudness in sones, all as functions of the grille face velocities.

The octave-band analysis serves the purpose of indicating the influence of certain variables on the frequency distribution. Loudness, an important criteria of grille noise, provides a standard whereby the test grilles can be placed in some rank and order with respect to an observer.

The chart shown in Fig. 2, devised by the Armour Research Foundation, provided a convenient tool for studying the loudness characteristics of grille noise. By plotting the octave-band data on this chart, the loudness spectra in sones were obtained. The method presented by this chart is based on the psychoacoustic experiments conducted by S. S. Stevens of Harvard University[1], and is now widely accepted over the older equivalent tone method.

TEST EQUIPMENT AND INSTRUMENTATION

Fig. 3 shows the air supply system used to obtain the required grille noise data. This system met the basic need for a controlled air supply having a low noise level, and an enclosed space of low ambient noise level where a wide range of grille noise intensity could be measured.

The air supply fan was a centrifugal type, single inlet, single outlet, having a 9-in. wheel. It was powered by direct-drive d-c motor, whose speed could be varied by means of a rheostat on an adjacent control board. A glass fiber filter was mounted on the inlet side of the fan, with a sheet metal transition, to prevent dirt from being deposited inside the system.

The air measurement duct contained an air-straightener section, Pitot tube, and thermometer connections, for determining the air flow rates to the test grilles. A

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† This paper is based on the author's M.Sc. thesis written for the University of Tennessee.

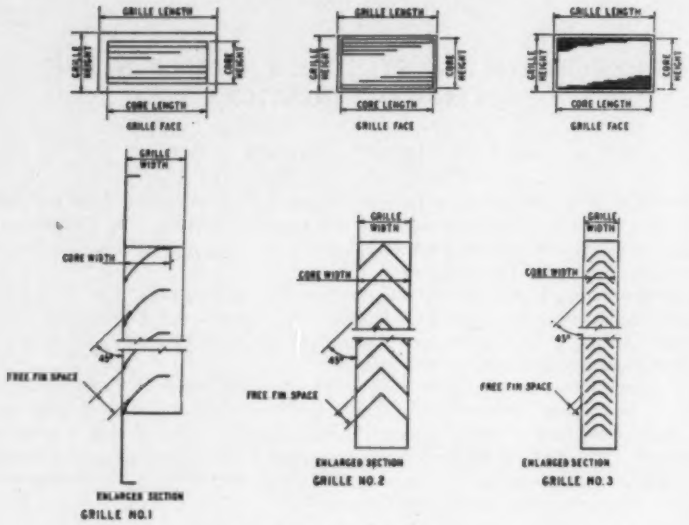
* Mechanical Engineer, Division of Design, Tennessee Valley Authority.

[1] S. S. Stevens: Measurement of loudness (*Journal Acoustical Society of America*, Vol. 27, September 1955, p. 815).

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N. Y., June 1959.

hook gage was connected to the Pitot tube, to obtain the velocity pressures, particularly at low flow rates.

A sound trap was required to reduce the noise intensity generated by the supply fan before it reached the test grilles. This trap, constructed of plywood, consisted of a series of acoustically-lined 180 deg turns. The first three sections of the trap had two air flow paths, and the last section had 4 air flow paths. The acoustic insulation used was 1-in. flexible duct liner. All of the insulation surfaces exposed



(See Table 2 for physical data)

FIG. 1. . . .GRILLES USED FOR EXPERIMENT

to the air flow were painted to prevent fiber corrosion, and reduce the surface friction factor.

The anechoic chamber, which measured 6 ft-6 in. wide x 7 ft-6 in. deep x 7 ft-0 in. high overall, provided the space for measuring the grille noise intensities. The typical wall construction consisted of glass fiber wedges on the inside, 4-in. thick insulation on the outside, with a 4-in. air space between. The wedges, 8 x 8 in. at the base and 10 in. high, were placed in an alternating pattern. The sound transmission loss through the wall was quite satisfactory for eliminating outside noise. Preliminary tests made when this chamber was first constructed indicated good free-field characteristics for frequencies above 300 cps.

One wall of the chamber was hinged to provide access to the inside. A sheet metal sleeve for holding the test grilles extended through another wall. On the opposite wall a number of wedges were removed to allow the air to exit through an exhaust duct to the outside of the building. This duct, which was constructed

of plywood, consisted of two airflow paths lined with 1-in. flexible duct liner. A static pressure tap was added to the grille mounting sleeve, for measuring the static-pressure loss through the test grilles.

The instrumentation for the noise measurements consisted of a microphone, sound-level meter, and an octave-band analyzer. The microphone was located at the center of the test grille, and 12 in. from it. Although this location avoided the

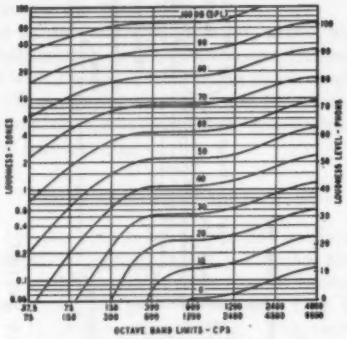


FIG. 2. . . .LOUDNESS CALCULATION CHART. TOTAL LOUDNESS EQUALS NUMBER OF SONES IN LOUDEST BAND PLUS 0.3 THE SUM OF SONES IN OTHER BANDS

direct discharge of air against it, the microphone was shielded with a suitable wind screen made of a wire frame covered with muslin.

TEST PROCEDURES

The basic information needed to correlate experimental data involved establishing for each grille the air flow rate, static pressure loss, total noise with grille in place, and background noise.

The air flow rates to each test grille were determined as a function of fan speed, using an 8-point Pitot tube traverse. A sufficient number of fan speeds were used to provide the calibration curves of air flow rate as a function of fan speed.

The first series of noise measurements were taken with the grille in place, representing the total noise. The octave-band sound-pressure levels, and grille static-pressure losses were determined for 8 fan speeds ranging from 800–1500 rpm. In addition, the velocity pressure reading at the midpoint of the air flow measurement duct was obtained for each test speed by means of the Pitot tube and an inclined draft gage. This pressure was used as a reference point to represent the air flow rate at the particular test speed.

The background noise at the microphone location was simulated by removing the test grille and supplying the same air flow rate, at the same test speeds, as were

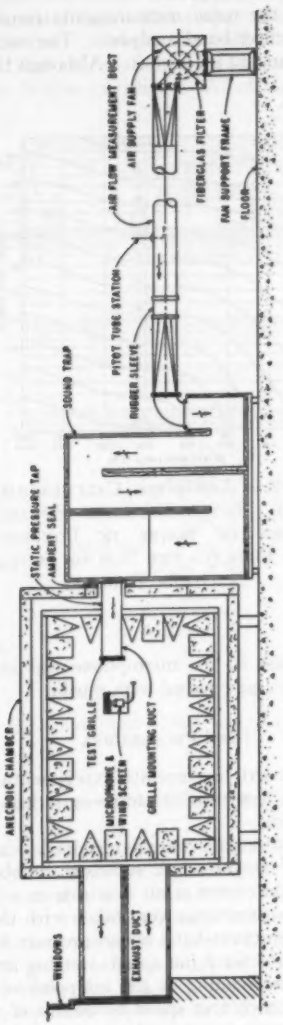


FIG. 3. . . AIR SUPPLY SYSTEM

used with the grille in place. In this manner fan and air turbulence noise could be reproduced at the microphone.

In order to provide the same flow rate for a given test speed, it was necessary to throttle the air flow system until the representative velocity-pressure reading at the midpoint of the air flow measurement duct was indicated. The necessary throttling was accomplished by adding a well-insulated enclosure around the filter housing at the fan inlet. The enclosure contained an opening fitted with a sliding panel which served as the throttling device, and reduced effectively the throttling noise before the air entered the system.

Generation of noise at the microphone wind screen was considered to be an important source of background noise at the lower frequencies. To produce the same air flow patterns with the grille removed, a nylon cloth deflector was attached to the outlet of the grille mounting duct. Using the same fan speeds and related velocity-pressure readings, the background noise octave-band sound-pressure levels were determined at the test location.

EXPERIMENTAL DATA

Experimental data for the octave-band pressure levels of the grille noise were corrected to obtain the sound-pressure levels per square foot of core area.

The total sound-pressure levels, SPL_t were first corrected for the existing background sound-pressure levels, SPL_b. The method that is usually correct is to subtract SPL_b from SPL_t on an energy basis. The number of decibels representing the sound-pressure levels can be converted to relative powers, since sound pressure is usually proportional to the square root of sound power. If W/W_o is the power ratio for a given number of decibels, and P/P_o is the corresponding pressure ratio, then

$$DB = 10 \log_{10} \frac{W}{W_o} = 10 \log_{10} \frac{P^2}{P_o^2}$$

If the value of P_o, the reference pressure, is 0.0002 microbars, then the equation defines the decibel unit to indicate sound-pressure level. Thus, SPL_e, the difference between total and background sound-pressure levels of grille noise is determined by taking the difference of the respective power ratios,

or
$$SPL_e = 10 \log_{10} \left[\text{antilog} \frac{W_t}{W_0} - \text{antilog} \frac{W_b}{W_0} \right], \text{ db}$$

or
$$SPL_e = 10 \log_{10} \left[\text{antilog} \frac{P_t^2}{P_o^2} - \text{antilog} \frac{P_b^2}{P_o^2} \right], \text{ db}$$

and
$$SPL_e = 10 \log_{10} \left[\text{antilog} \frac{SPL_t}{10} - \text{antilog} \frac{SPL_b}{10} \right], \text{ db}$$

For example, if the measured SPL_t for a particular octave-band is 85 db, and the existing SPL_b in that band is 80 db, then SPL_e, the noise attributed to the grille is

$$SPL_e = 10 \log_{10} \left[\text{antilog} \frac{85}{10} - \text{antilog} \frac{80}{10} \right] = 83.3 \text{ db}$$

Background noise correction was required in the octave-bands below 300 cps. The higher background noise in the lower octave-bands can be attributed to the

predominance of low-frequency noise generated at the microphone wind screen, and to the poor efficiency of the sound trap in attenuating low-frequency fan noise. The correction at the higher octave-bands was essentially equal to zero.

In order to compare the noise levels of each grille on an equal basis, it was necessary to correct SPL_c to obtain SPL_s, grille noise per square foot of core area. Similar to the background noise correction, the area correction was determined on

TABLE NO. 1. . . . DECIBEL CORRECTION FOR GRILLE CORE AREA

GRILLE NO.	A_g, SQ. FT.	10 $\log_{10} A_o/A_g$ (db)
1	0.467	3.30
2	0.623	2.04
3	0.646	1.90

an energy basis. Thus, SPL_s can be represented by acoustic power per square foot, as follows:

$$\text{Antilog } \frac{SPL_s}{10} = [(W_s/W_o)/(A_g/A_o)]$$

where

A_o = one square foot.
A_g = grille core area, square feet.

therefore

$$SPL_s = 10 \log_{10} [(W_s/W_o)/(A_g/A_o)], \text{ db}$$

$$SPL_s = 10 \log_{10} \frac{W_s}{W_o} + 10 \log_{10} \frac{A_o}{A_g}, \text{ db}$$

$$SPL_s = 20 \log_{10} \frac{P_s}{P_o} + 10 \log_{10} \frac{A_o}{A_g}, \text{ db}$$

$$SPL_s = SPL_c + 10 \log_{10} \frac{A_o}{A_g}, \text{ db}$$

The decibel correction for area to be added to SPL_c is 10 $\log_{10}$ (A_0/A_g). For example, a 0.5 sq ft grille core, with a SPL_c of 83.3 db for a specific octave-band, would have an area correction of 10 $\log_{10}$ (1/0.5) or 3 db. The resulting SPL_s for the core would be 86.6 db. The same result is obtained by adding the SPL_c for two 0.5 sq ft grille cores. The area correction values for each test grille are tabulated in Table 1.

The resultant grille noise data, SPL_s, are presented graphically as a function of grille face velocity for each octave band as shown in Fig. 4. The curves for Grille No. 1 are extrapolated to the lower velocities, as shown by the dashed portion, in order to allow a wider area of comparison. This extension seems reasonable when considering the general straight-line trend of the curves.

The grille face velocities were determined from the air flow rates and the test grilles free core area tabulated in Table 2. The free core areas were measured on a plane perpendicular to the air flow discharge at the grille face, which was taken as 45 degrees for all the grilles.

The octave-band spectra for each grille, at various grille face velocities, shown in Fig. 5, were plotted from the smoothed curves of Fig. 4. By combining the octave-band spectra with the Loudness Calculation Chart, the loudness spectra for each grille were obtained, as shown in Fig. 6. The total loudness curves for

NOMENCLATURE

A = core area, square feet.
a, b = constants dependent upon the grille and octave band of interest.
db = abbreviation representing decibels.
P = sound pressure, microbars.
SPL = octave-band sound-pressure level in decibels relative to 0.0002 microbars.
V_f = average grille-face velocity in a plane perpendicular to the air flow discharge at the grille face, feet per minute.
W = sound-power, watts.

Subscripts

$_b$ = background noise.
$_c$ = corrected for background noise.
$_g$ = grille.
$_o$ = reference.
$_s$ = grille noise per square foot of core area.
$_t$ = total noise.

TABLE 2. . . .PHYSICAL CHARACTERISTICS OF TEST GRILLES

Grille Number	1	2	3
Grille length, inches	14	14	14
Grille height, inches	8 3/8	8	8
Grille width, inches	1 1/16	15/16	5/8
Core length, inches	11 1/2	12 15/16	13 1/16
Core height, inches	5 27/32	6 1/16	7 1/8
Core width, inches	7/8	15/16	1/2
Grille area, square feet	0.813	0.778	0.774
Core area, square feet	0.467	0.623	0.646
Free core area, square feet	0.256	0.399	0.401
Free core area/Core area	0.547	0.640	0.620
Number of fins	8	15	40
Fins per inch	1.4	2.2	5.6
Fin thickness, inches	0.030	0.030	0.015
Number of free fin spaces	7	16	41
Free fin space, inches	0.457	0.278	0.108

each grille shown in Fig. 7 were evaluated from their respective loudness spectra and the summation procedure shown in Fig. 2.

Again, the graphical presentations for Grille No. 1 were extended to the lower grille face velocities for comparison purposes, as shown by the dashed curves in Figs. 5, 6, and 7.

GRAPHICAL RESULTS

From the graphical presentation shown in Fig. 4 a straight-line relationship is established between the octave-band sound-pressure levels of grille noise and the

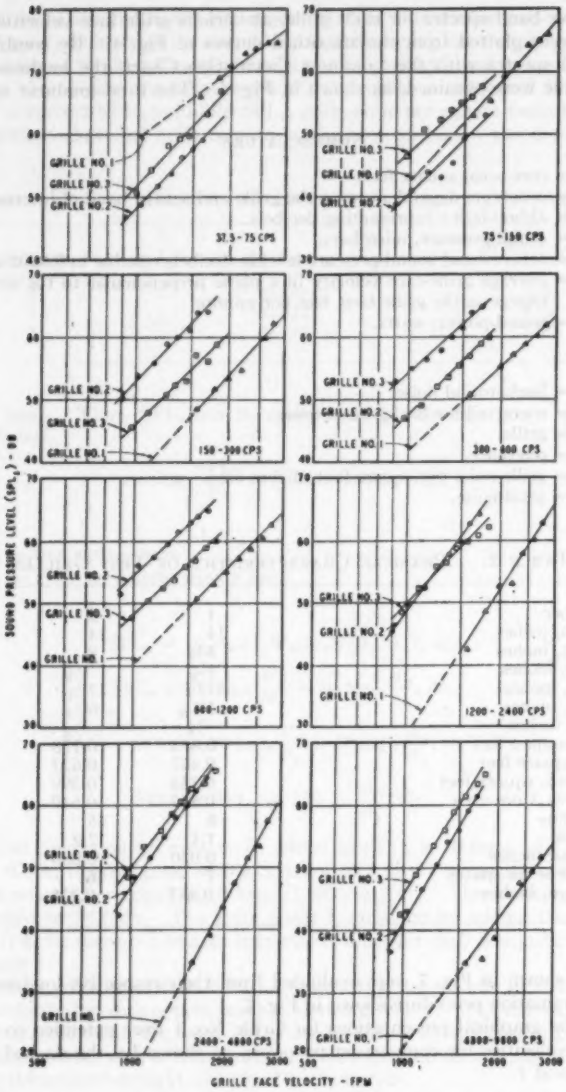


FIG. 4. . . .COMPARISON OF SOUND-PRESSURE LEVELS OF GRILLE NOISE FOR EACH OCTAVE BAND

logarithms of the grille face velocities. This relationship is expressed by the following equation:

$$SPL_s = a \log_{10} V_f + b, \text{ db}$$

where

a and b = constants dependent upon the grille and octave-band of interest.
V_f = grille face velocity, fpm.

Deviations from the straight-line appear mainly in the octave-bands below 300 cps. Apparently, these deviations can be attributed to the difficulties of measuring the background noise in the lower octave-bands.

Fig. 4 also shows that the slope of the curves increase in the higher octave-band. This would indicate that at the higher frequencies, grille noise increases at a more rapid rate with increasing velocities.

In Fig. 5 which shows the octave-band spectra for each grille, it is observed that as the grille face velocity increases, the spectra contain greater percentages of the higher frequencies. This results in greater predominance of the higher octave-bands in the case of Grilles Nos. 2 and 3. A comparison of the frequency distribution for the grilles indicates that some correlation exists between the number of fins in the grille core and the predominant octave-band produced. Thus, Grille No. 1 with 1.4 fins per in., has predominance in the lowest frequency band; Grille No. 2, with 2.2 fins per in. has the greatest percentage in the middle frequency band; and Grille No. 3 with 5.6 fins per in. has the greatest percentage in the higher frequency bands.

From Fig. 5, it is evident that the sound-pressure level magnitudes for Grilles Nos. 2 and 3 are greater, as compared to Grille No. 1, mainly in the higher-frequency bands. The greater acoustic energy contained by the noise of Grilles Nos. 2 and 3 is an indication of their greater aerodynamic losses due to air flow, which can be represented by the static-pressure loss curves shown in Fig. 8.

The differences in pressure losses for the 3 test grilles reflect the differences in fin design. The fin spacing, shape, depth, and thickness are important factors which affect, to varying degrees, the static-pressure losses. It is apparent that more fins per inch result in higher static-pressure losses due to more friction surfaces. This fact is substantiated by the higher pressure losses for Grilles Nos. 2 and 3.

Although Grille No. 2 has less fins per inch than Grille No. 3, its pressure losses are higher. The reasons for this are seen in Fig. 1. The fin shape for Grille No. 2 is a sharp 90-degree bend as compared with the rounded fin shape for Grille No. 3. In addition the linear distance of air flow through the fins for Grille No. 2 is longer as compared to Grille No. 3.

Although Grille No. 1 has the smallest percentage of free core area, its static-pressure losses are much lower. The reasons for this can be attributed to Grille No. 1's smaller number of fins per inch, and the smooth 45-degree bend of the fins.

In Fig. 6 the loudness spectra of noise for each grille are shown. It is apparent that the sound-pressure level magnitudes in the higher-frequency bands are most important. In addition, as the grille-face velocity increases, the higher-frequency bands become louder. This is due to the greater percentage of the higher frequencies occurring at the higher grille-face velocities.

The total loudness curves for the grille noise presented in Fig. 7 summarize the loudness spectra. The calculation method used for total loudness indicates that

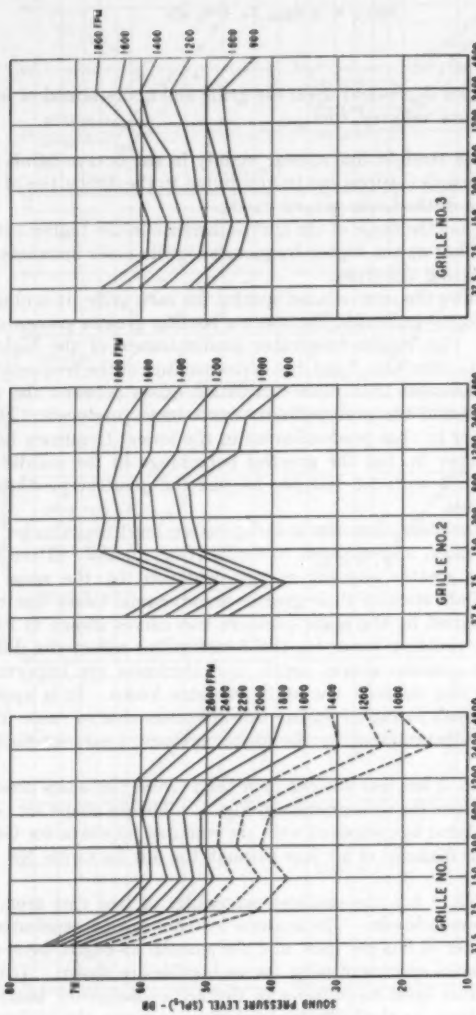


FIG. 5. . . . OCTAVE-BAND SPECTRA OF GRILLE NOISE AT VARIOUS GRILLE-FACE VELOCITIES

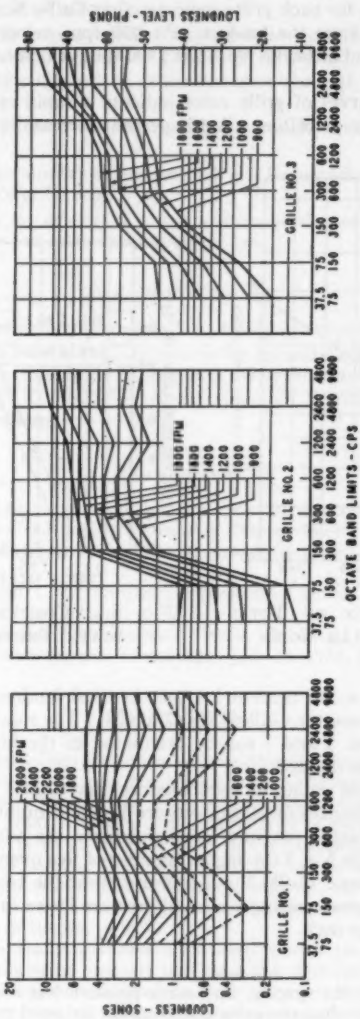


FIG. 6. . . LOUDNESS SPECTRA OF GRILLE NOISE AT VARIOUS GRILLE-FACE VELOCITIES

these curves mainly represent the loudest octave-bands. It is important to observe that the strong low-frequency octave-bands that appeared for Grilles Nos. 1 and 3 had but little effect on the total loudness of the grille noise. A comparison of total loudness for each grille indicates that Grille No. 1 is the quietest, and Grilles Nos. 2 and 3 are the loudest. At 900 fpm velocity, Grilles Nos. 2 and 3 are 2.7 times as loud as Grille No. 1; at 1800 fpm velocity, they are 3.2 times as loud.

The total loudness curves of grille noise indicate a rapid rise of the loudness with increasing grille-face velocities. A 100 percent increase in velocity from 900

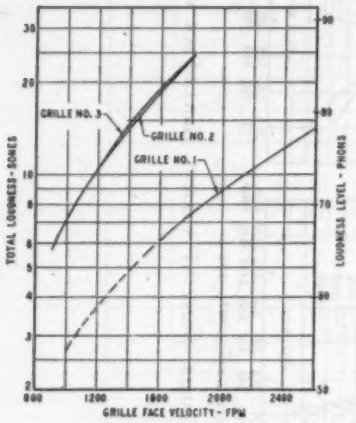


FIG. 7. . . .COMPARISON OF TOTAL LOUDNESS OF GRILLE NOISE

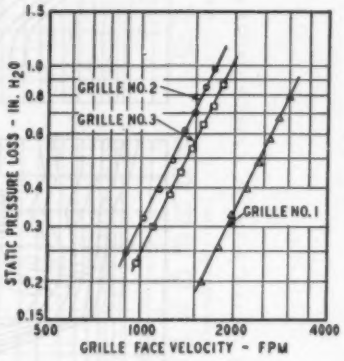


FIG. 8. . . .COMPARISON OF GRILLE STATIC PRESSURE LOSSES

to 1800 fpm results in a 275 percent increase in total loudness for Grille No. 1, and a 340 percent increase for Grilles Nos. 2 and 3. The more rapid rise in total loudness for Grilles Nos. 2 and 3 can be attributed to the higher-frequency predominance in their noise spectra.

Observe that the equal values of total loudness for Grilles Nos. 2 and 3 point out the cumulative influences of static-pressure loss and number of fins per inch. For Grille No. 2, the higher pressure losses overcome the advantage of its small number of fins; for Grille No. 3 the higher number of fins overcome the advantage of smaller pressure losses. Grille No. 1 indicates that the best fin characteristics for producing a low loudness rating are small pressure losses in combination with a small number of fins per inch.

In general

1. Grille-face velocity, fin spacing, and static-pressure loss are interrelated factors influencing the frequency characteristics and loudness rating of grille noise.

2. Octave-band sound-pressure level of grille noise is a function of the grille face velocity which can be expressed as:

$$SPL_s = a \log_{10} V_f + b, \text{ db}$$

3. Increasing the grille-face velocity results in a greater percentage of higher frequencies in the grille noise spectrum.

4. Grilles with a larger number of fins per inch tend to have higher-frequency predominance in the grille noise spectrum.

5. Grilles with higher pressure losses have greater magnitudes of sound-pressure levels in the high-frequency range of the grille noise spectrum.

6. Loudness rating of grille noise is dependent upon the high-frequency range of the grille-noise spectrum.

7. Increasing the grille face velocity results in a rapidly increasing loudness rating of the grille noise.

8. Grilles with a large number of fins per inch or a large static-pressure loss, will have a high loudness rating of grille noise.

9. Best combination for producing a low loudness rating of grille noise is a small number of fins per inch and a low static-pressure loss.

ACKNOWLEDGMENTS

The author expresses appreciation to P. F. Pasqua, University of Tennessee, for his encouragement and helpful suggestions; to the following companies for providing the test grilles used in this study: Titus Mfg. Co.; Arolite Co.; and the Barber-Colman Co.; and to F. G. Tyzzer, Armour Research Foundation, for use of the Loudness Calculation Chart.

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DISCUSSION

W. T. KERKA, Cleveland, Ohio (WRITTEN): The author has written a very informative paper, but it is felt that he has omitted in his analysis, the attenuation effect of the grilles. In his paper, Mr. Marvet has given an example illustrating the method of determining grille noise; that is, with a background level of 80 db (without the grille in place) and with a level of 85 db with the grille mounted in position, the contributing noise by the grille would be 83.3 db. This would be true only if the grille did not attenuate any of the input signal to it. Actually, however, the grille does attenuate sound, the maximum attenuation occurring at a frequency where the wavelength is about 4 times the width of the fin.

What the author has actually measured is the input level to the grille and the output level from it. Taking the same example and assuming that the grille attenuates the input signal by 2 db, the attenuated signal reaching the microphones would be 80 − 2 = 78 db, if the grille generated no noise. With noise generation, however, the output signal is 85 db, hence the level contributed by the self-generated noise of the grille would be 84 db instead of 83.3 db. In the author's case, this discrepancy would apply to the levels below 300 cps only, since above 300 cps the background level was low enough not to contribute anything to the generated level of the grille.

The procedure for determining the generated levels would be carried through for each of the octave-bands. The attenuation values could be found by using a speaker sound source. Since the grilles are non-dissipating in acoustic character, the attenuation values found under no-flow conditions would hold true with air flow.

W. W. KENNEDY, Rockford, Ill., (WRITTEN): The paper was most interesting, and there are several comments.

The 3 grilles used in the test are indicated to be of the usual return-air type. Actually only one of the grilles (No. 1, Fig. 1) is designed for return-air service, having widely spaced fins for low resistance and minimum disturbance of the air stream. The 2 other grilles used (Nos. 2 and 3, Fig. 1) are designed for the transfer of air between adjacent areas where sight proof characteristics are desirable. They obviously would have quite different pressure loss and noise characteristics than the return type grille.

The face velocities used in the test are much higher than found in normal practice. The velocity through the transfer grille is a function of the pressure difference between the areas served which generally limits the flow to less than 500 fpm.

HARRY KINNEY*, Detroit, Mich.: Did the author make any studies with regard to the grilles in the 35-degree angle? Some suppliers of grilles are now offering that angular type. In writing fairly open specifications, we are specifying both the 35- and 45-degree angles on grilles.

AUTHOR'S CLOSURE: The author agrees with Mr. Kerka that the grille attenuation effect should be considered, assuming that the input level, or background noise, originates only in the air supply path to the grille. Actually the background noise levels measured included the low-frequency noise generated at the microphone windscreen. It was not determined whether this source of background noise, or the air supply background noise was predominant. Actual observation of the air impingement upon the windscreen during the background noise test indicated that this noise source was quite evident. The importance of the grille attenuation would be dependent upon which of the background noise sources was predominant. The author feels that this grille attenuation error would not affect the relative differences of the grille noise characteristics as shown graphically.

In answer to Mr. Kinney, the author did not make any study of grilles having a fin shape angle, other than shown in Fig. 1. The fin angle is important with respect to its effect upon the core design, and the resulting differences in number of fins per inch, and static-pressure losses, which in turn will affect the grille noise characteristics. It is hoped that further study of this subject will consider the direct influence of fin angle variations upon grille noise characteristics.

Mr. Kennedy is correct in stating in his comments that Grilles Nos. 2 and 3 are not the usual return-air types. The author has considered these grilles as a special case of the return-air type. There are some situations in an air-conditioning distribution system where this type of grille is desirable for returning air. For example, this type of grille, with its light-proof characteristics, was placed in the door of a dark room, allowing return air to pass into the adjacent corridor as required. The three test grilles used in this study were selected because of the related shapes of the fins, and the differences in fin spacing and pressure losses, rather than the practical applications of such grilles.

As was noted by Mr. Kennedy, the face velocities used in this study were much higher than found in normal practice. The higher velocities were used as a necessity of the problem in order to obtain sound-pressure level readings well above the ambient level in the test chamber. It should be noted that the sound-pressure levels reported in this study are valid only for the test setup used. The purpose of the experimental data was to predict the characteristic patterns inherent in grille noise.

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* Harley, Ellington, and Day, Inc.

1690

No. 1690

EFFECT OF FUEL COMPOSITION ON OIL BURNER NOISE

By W. A. BEACH*, R. W. SAGE**, AND H. F. SCHROEDER†, LINDEN, N. J.

THE IMPORTANCE of noise to the purchaser of heating and air-conditioning equipment has been amply demonstrated. This Society has devoted considerable effort toward understanding and solving the problems associated with noise. Much of this effort has been directed at the problems of mechanical noise such as compressor and blower operation, duct noise, and insulation. In oil-fired furnaces and boilers, however, much of the noise is non-mechanical in its origin and is, in fact, due to the flame itself. This noise is frequently referred to as combustion *roar*.

There has been some feeling that fuel composition contributes heavily to the intensity of flame noise. In particular, at the June, 1957, meeting of the ASHAE in Murray Bay, Que., there was considerable discussion of factors that influenced flame noise††. The discussions indicated divergent views on the effect of fuel composition, in this case No. 2 heating oil, in contributing to the amount of noise produced. Because data to answer this question were not readily available, an investigation was carried out to evaluate the effect of fuel composition on domestic oil burner noise.

Flame noise of two basic types is recognized. These are the normal combustion *roar* and occasionally a high amplitude *throbbing* known as pulsation. A variety of fuels were tested at both conditions using conventional burners.

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* Process Research Division, Esso Research and Engineering Co.
** Section Head, Process Research Division, Esso Research and Engineering Co.
† Process Research Division, Esso Research and Engineering Co.
†† Discussion, ASHAE TRANSACTIONS, 1957, p. 425.
Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N. Y., June 1959.

TYPICAL FURNACE BURNER COMBINATION CHOSEN

A conventional warm air furnace with an output rating of 72,000 Btu per hr was used for the test work. It is representative of the smaller sized furnace which is gaining popularity in new homes today. The oil burners tested in this furnace were also typical of those found in the field. A commercially available high-

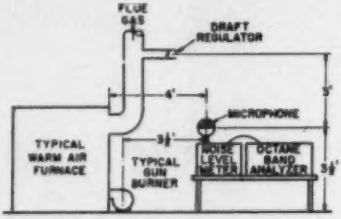

FIG. 1. . . .MICROPHONE ORIENTATION USED IN TESTS

pressure gun burner was used to study normal combustion noise; for pulsation flame noise, a typical low pressure air atomizing burner.

The combustion chamber was a 9-in. diam preformed chamber provided by the furnace manufacturer as an integral part of the unit. It was made of hard firebrick and is similar to the type installed in many modern heating plants.

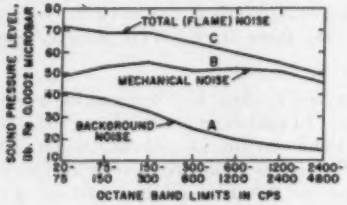

FIG. 2. . . .CUMULATIVE NOISE LEVELS BUILT UP FROM THREE COMPONENTS

All variables were held constant throughout the test except for fuel composition. The firing rate was adjusted for each fuel and burner to provide a constant heat input of 90,000 Btu per hr which is approximately equal to 0.65 gph. The quantity of combustion air was adjusted for each case to be 30 percent in excess of stoichiometric requirements. With No. 2 heating oil this was the setting required for zero smoke. For some of the other fuels, however, this setting caused considerable smoke formation. Draft was maintained at 0.015 in. H_2O measured over the fire.

ACCURATE INSTRUMENTS USED TO EXAMINE ENTIRE SOUND SPECTRUM

Burner noise is composed of sound waves of all frequencies in the audible spectrum (20–20,000 cps). However, in most test work, it is not necessary to analyze the spectrum for sound-pressure level as a continuous function of frequency, rather, the sound level is measured in *octave bands*. These octave bands have the

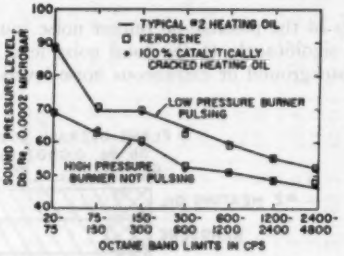

FIG. 3. . . .SOUND-PRESSURE-GRADIENTS FOR THREE FUELS

same frequency limits as octaves on the piano keyboard. The sound-pressure level gradient obtained by this method gives a good approximation of the entire spectrum and was used in this analysis of burner noise.

TABLE 1. . . .SPECIFICATIONS OF FUELS TESTED

	GRAVITY ° API	COMPOSITION, WT. PERCENT				BOILING RANGE, F[a]		
		PARAFFIN	NAPHTHENE	OLEFIN	AROMATIC	10%	50%	90%
No. 2 Heating Oil	35.5	37.1	29.9	2.3	30.7	396	478	562
Kerosene	41.2	39.6	41.6	0.6	18.2	382	434	487
100% Cat. Cracked Heating Oil	30.0	—53.4—		12.6	34.0	355[b]	465	610[b]
100% Thermally Cracked Heating Oil	36.4	—37.7—		28.8	33.5	330[b]	442	590[b]
Methyl Dicyclopentadiene	19.3	0.0	0.0	100.0	0.0	330	330	330
Benzene	28.6	0.0	0.0	0.0	100.0	176	176	176
n-Cetane	51.1	100.0	0.0	0.0	0.0	550	550	550
n-Heptane	74.2	100.0	0.0	0.0	0.0	209	209	209
Iso-octane	71.8	100.0	0.0	0.0	0.0	211	211	211

[a] Appropriate ASTM Distillations.
[b] Initial and final boiling points.

The sound instruments used in the program gave results reproducible within 1 db. Under normal conditions the human ear can barely detect this one decibel change in sound-pressure level. A commercially available sound-level meter was

used for detection, and sound measurements in the octave bands were made with a standard octave-band frequency analyzer. These instruments were checked and calibrated daily. They were always located in the same position relative to the furnace and this location was chosen as the point at which the sound-pressure level was most critical. The microphone orientation is shown in Fig. 1*.

SOUND MEASUREMENT REVEALED THE FLAME TO BE THE MAJOR NOISE SOURCE

Sound measurements of the possible oil burner noise sources showed that only the flame contributed significantly to the total noise level. Three noise sources were considered, (1) background or extraneous noise levels, (2) mechanical noise

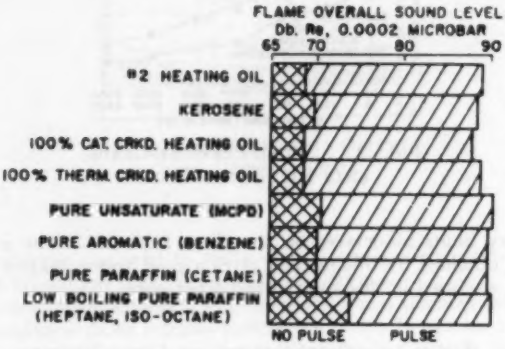

FIG. 4. . . .TEST RESULTS WITH VARIOUS FUELS

from the burner and furnace, and (3) the flame itself. Fig. 2* shows the cumulative noise levels built up from these 3 components.

The background noise level, shown as Curve A of Fig. 2 was measured with the burner and furnace turned off. Since the sound laboratory was remotely located, it had an extremely low background noise level, the equivalent of a radio broadcasting studio. The sound-pressure level (SPL) gradient obtained was at least 30 db below the total burner noise in all octave bands. Since the background noise level was so much less than the burner noise it did not contribute to the total noise level. This is due to the logarithmic basis of the decibel sound-level scale which means that decibel readings cannot be added arithmetically. For example, if two noise sources have individual sound levels of 80 db, the total sound level when running together would be 83 and not 160 db. As the individual sound levels become further and further apart, the combined level becomes essentially that of the higher level noise source.

The mechanical noise level shown as Curve B of Fig. 2 was measured with all the mechanical components of the burner and furnace operating. This noise level included the furnace blower, oil pump, motor, air fan and ignition system. The mechanical noise level was also sufficiently below the total burner noise level to

* In the figure where the word octane appears, the word should be octave.

TABLE 2. . . .COMBUSTION NOISE FROM VARIOUS FUELS

	SOUND-PRESSURE LEVEL IN EACH OCTAVE BAND, DB RE 0.0002 MICROBARS									REMARKS
	OVER-ALL (20-20K)	20-75	75-150	150-300	300-600	600-1200	1200-2400	2400-4800	4800-9600	
No. 2 Heating Oil	89.0	89.0	69.0	69.0	64.5	59.5	55.5	52.0	40.0	Low pressure burner with pulsing flame.
Kerosene	88.5	88.5	70.0	70.0	63.0	59.0	55.0	53.0	41.0	
100% Cat. Cracked Heating Oil	88.0	88.0	69.0	68.0	63.5	59.0	55.0	52.0	40.5	
100% Thermally Cracked Heating Oil	89.0	89.0	69.0	68.5	64.0	59.5	55.0	52.0	40.0	
Methyl Dicyclopentadiene	90.5	90.5	71.0	69.5	66.0	62.0	56.5	50.5	40.0	Low pressure burner with pulsing flame.
Benzene	90.0	90.0	70.5	69.5	65.5	61.5	55.5	50.0	39.0	
n-Cetane	90.0	90.0	70.5	71.5	66.0	62.0	———	———	———	
n-Heptane	90.0	90.0	76.0	73.0	67.0	64.0	60.0	57.5	48.0	
Iso-octane	90.5	90.5	72.5	72.5	65.5	62.5	58.5	54.0	44.0	
No. 2 Heating Oil	69.0	67.5	62.0	58.0	52.0	51.0	48.5	47.0	40.5	High pressure burner with nonpulsing flame.
100% Cat. Cracked Heating Oil	69.0	67.5	63.0	57.5	51.5	51.0	48.5	48.0	41.0	
Kerosene	69.0	68.0	62.5	58.0	52.5	51.5	48.5	48.5	40.0	

render it a minor noise contributor. Since the only other noise source was the flame itself, it is obvious that the flame was the major source of noise in this oil burner. Curve C of Fig. 2 therefore represents both the total noise level and the flame noise level.

FUEL COMPOSITION DOES NOT AFFECT FLAME NOISE

Nine different fuels were tested for their effect on the largest oil burner noise contributor, the flame. These test fuels included typical products now marketed, fuels composed entirely of cracked products and several pure compounds. The properties of these fuels are listed in Table 1.

Despite wide variations in fuel composition, there were no significant differences in the overall noise level or in the noise level in any of the individual octave bands with fuels boiling in the range of kerosenes and No. 2 home heating oils. A fuel composed of 100 percent catalytically cracked oil, and kerosene, a straight run virgin product, gave the same sound spectrum as regular No. 2 heating oil in both the low pressure and high pressure gun burners. These SPL gradients are shown in accompanying Fig. 3*.

The results were independent of the type burner used. Although the high pressure burner SPL gradient is inherently lower than the low pressure burner SPL gradient, it is, nevertheless, similarly insensitive to fuel changes. Even the highly unsaturated methyl dicyclopentadiene, which can be considered an extreme of possible cracked oils, did not vary from the noise level of No. 2 heating oil. Normal cetane, which is a pure paraffin boiling at 550 F was also similar to No. 2 heating oil.

Of all the tests made, only the lowest boiling paraffins, normal heptane and isooctane, showed noise levels that varied significantly from the other fuels tested. These pure compounds are found in gasolines and are not used as fuels for regulation type oil burners. The low boiling benzene, an aromatic, had the same noise level as No. 2 heating oil. These test results are shown graphically in Fig. 4*.

Even here, however, the differences in noise level were not large except towards the higher frequency end of the spectrum. This can be seen in Table 2. While no verified reason is known for this somewhat greater noise level, it may be associated with higher volatility of these compounds which, in turn, means the fuel burns faster in the combustion zone when they are introduced there.

CONCLUSION

The conclusion is that the oil burners used in home heating equipment are not sensitive to fuel composition from a noise level point of view. The extreme ranges of compositions tested are far beyond the limits which any refinery product could cover. Even the extremes, however, gave essentially the same noise levels. Changing fuel composition, therefore, does not appear to be a method by which oil burner noise can be reduced.

DISCUSSION

D. W. LOCKLIN, Columbus, Ohio: This paper fills an important gap in the literature on the combustion of distillate fuels.

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*In the figure where the word octane appears, the word should be octave.

In any investigation of this type it is necessary to fix some of the important variables. The authors have chosen to fix Btu input and excess air. Similarly, a case could be made for controlling some other combination of variables such as volumetric firing rate, air-shutter opening, or smoke level. Frequently, combustion noise amplitude will be affected by the degree of excess air. However, another approach would probably not significantly alter the conclusions of the authors in the range of commercial distillate fuels.

During 1948 and 1949, the Oil-Heat Institute sponsored a combustion research program at the Research Laboratory of this Society for the purpose of developing a burner system which could be used for rating the burning qualities of fuel oils. Sound data were not recorded during this investigation, but test conditions where audible pulsation occurred were so noted. Examination of these data for widely differing fuels did not reveal any pattern of correlation with fuel properties, but pulsation depended more on the conditions of operation imposed upon the combustion system. Although no quantitative data on sound were recorded, these observations would tend to support the conclusions of the authors that differences in fuels have little effect on the presence of audible pulsation.

In Fig. 1, the location of the microphone is shown in proximity both to the burner and also to the barometric draft control. The draft condition, which would be the factor establishing the opening of the draft control was probably held constant during these tests. If so, the sound radiation characteristics of the control opening would be constant. Could the authors comment on that point?

At Battelle Memorial Institute this work has been followed with interest and it is pleasing that the authors have investigated this particular phase of combustion acoustics in oil-fired systems.

W. H. BOYLAN*, Trenton, N. J.: Does this noise program contemplate a study of air injection *vs* solid injection? My company used the air injection principle of oil burning and introduced compressed air in the nozzle rather than using the customary solid injection.

As far as combustion is concerned, was very little difference detected between air injection and solid injection?

C. D. POTTER, Providence, R. I.: There was a 90,000 Btu an hour input. What size combustion chamber and floor area was needed? Do the authors find by changing the area and volume of the combustion chamber that the noise is decreased?

AUTHORS' CLOSURE (Mr. Schroeder): The draft was held constant at 0.015 in. of water for all the tests. The opening of the draft control was therefore about the same for each case. By the way, subsequent tests at higher drafts did not alter the sound level significantly.

Also, just a word about the comment on excess air and smoke. The combustion air input was held constant for each case to be 30 percent in excess of stoichiometric requirements. All but three of the fuels burned with less than a three smoke number at this air setting.

The authors examined domestic burners both low pressure air atomizing, and conventional high pressure types. The mechanical noise level, which includes everything but the flame noise, was in general slightly higher for the low pressure burners. This higher noise level was probably caused by the air compressor on the pump. The air flowing through the nozzle contributes nothing to the total noise level.

With a pulsing flame the noise levels were exactly the same for air injection and solid injection. When not pulsing, the air atomizing burner had about the same noise level as some of the high pressure burners tested.

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* General Electric Company.

Concerning Mr. Potter's question, the furnace had an output rating of 72,000 Btu per hr. The firing rate of the burners were set for approximately 90,000 Btu per hr. to allow for the efficiency of the furnace. The 9-in. diam combustion chamber supply by the furnace manufacturer was used for the test. Subsequent tests with different combustion chamber sizes, shapes and materials showed very little difference in noise level. It should be pointed out that these tests were made using firing rates from 0.5 to 1.0 gal per hr. At higher rates, the effect of combustion chamber design might be more pronounced.

1691

No. 1691

COMPARISON OF FLUOROALKANE ABSORPTION REFRIGERANTS

By B. J. Eiseman, Jr.*, Wilmington, Del.

ABSORPTION refrigeration is, in principle, an attractive method for utilizing fuel directly for cooling and is being so used in practical applications at the present time. However, existing absorption systems suffer from certain deficiencies. The present study was undertaken with the objective of overcoming them.

Three types of system deserve practical consideration at present: those in which water is the refrigerant, those in which ammonia is the refrigerant, and those having a halogenated organic compound as the refrigerant.

The principal practical deficiency of water as the refrigerant is that a water-cooled-condenser system is required to reach even air-conditioning cooling levels and that refrigeration temperatures cannot be reached practically even with a water-cooled condenser. Other disadvantages of water, with salt solutions such as lithium bromide as absorbent, are its corrosive nature and the fact that extremely low pressures must be maintained, a condition which is made the more difficult to meet by corrosion.

While ammonia as the refrigerant (water as the absorbent) can be used with an air-cooled condenser, its toxicity, flammability and explosiveness when mixed with air must be considered. Moreover, the system is far from ideal in technical respects. In particular, water is too volatile as an absorbent, and the system has an undesirably high operating pressure.

Most of these deficiencies can be overcome by a suitable choice of a fluoroalkane refrigerant, since these compounds are operative with air-cooled condensers and, being nonflammable and practically nontoxic, make direct cooling of peopled areas permissible. The coefficient of performance will be adversely affected by going to higher condenser temperature (or lower evaporator temperature), other things remaining unchanged.

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* Freon Products Laboratory, E. I. duPont de Nemours & Co., Inc. Member of ASHRAE.

Presented at the Annual Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Lake Placid, N. Y., June 1959.

The foregoing evaluation will be amplified and completed after developing data on the performance and other characteristics of the halogenated alkane absorption refrigerants in the sections which follow.

PERFORMANCE OF HALOGENATED ABSORPTION REFRIGERANTS

The most commonly used refrigerant today is Refrigerant 12. That being the case, an explanation is in order as to why it is not proposed as an absorption refrigerant. The primary reason is that Refrigerant 12 (CCl_2F_2) and other fully halogenated refrigerants such as Refrigerant 11 (CCl_3F), Refrigerant 113 ($CCl_2F \cdot CClF_2$) and Refrigerant 114 ($CClF_2 \cdot CClF_2$) show only Raoult's law solubility or *normal* solubility in liquids which might be used as absorbents. What is wanted is highly abnormal, enormous solubilities, so that volumes and heat losses can be kept to a minimum. A more exact statement would be that what is required is the largest possible difference in solubility between absorber conditions and generator conditions (which turns out to be the case when the solubility is extremely high as discussed more fully in later paragraphs). Among the halogenated alkanes, those having hydrogen as well as halogen in the molecule show the highest solubilities in absorbent liquids.

Six fluoroalkanes have been studied. In order of decreasing boiling point they are: Refrigerant 21 ($CHCl_2F$), Refrigerant 133a ($CH_2Cl \cdot CF_3$), Refrigerant 31 (CH_2ClF), Refrigerant 124a ($CHF_2 \cdot CClF_2$), Refrigerant 134 ($CHF_2 \cdot CHF_2$), and Refrigerant 22 ($CHClF_2$). A nonfluorinated compound, Refrigerant 30 (methylene chloride, CH_2Cl_2) is also included, since it has been used as an absorption refrigerant[1, 2].

For a complete and detailed comparison of absorption systems, full, accurate information would be required on the heat capacity, vapor pressure, heat of vaporization, density, viscosity and thermal conductivity of the refrigerant, the absorbent and their solutions, together with their heat of mixing. Data also would be required on the rate of solution and disengagement of the refrigerant and the rate of heat transfer under the unit operating conditions. Such information is not completely available. However, it has proved feasible to make an adequate comparison of refrigerants on the basis of more limited data.

For this purpose, the published engineering data of G. F. Zellhoefer[3] for a Refrigerant 21 unit, using the dimethyl ether of tetraethylene glycol as absorbent, have been used as a standard of comparison. The comparison of the six other refrigerants with Refrigerant 21 has been made on a ratio basis, so that the relative merits of the refrigerants are established independently of the particular machinery used in Reference 3. That is, any improvement or change in such factors as spraying or heat transfer should affect all of the refrigerants similarly, leaving their relative order of performance unchanged. The dimethyl ether of tetraethylene glycol was used by Zellhoefer as the best absorbent available. His choice has been confirmed by these tests.

The cycle temperatures are as follows: evaporator, 40 F; absorber 95 F; generator, 232 F; condenser, 104 F. The calculation is given step by step in Table 1. Additional explanation is needed regarding only the following steps:

Step (14). The amount of refrigerant is that required to effect 200 Btu per min of external cooling at 40 F and to cool the refrigerant itself as liquid from 104 to 40 F.

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[1] Exponent numerals refer to References.

Step (*37*). Total heat requirements could be calculated directly only for Refrigerant 21 and Refrigerant 22 since the data are incomplete for the other refrigerants. Comparison of this value for Refrigerant 21, Refrigerant 22 and Refrigerant 124a (only a partial calculation could be made for Refrigerant 124a) with the heat required to raise the weak liquor from 95 to 232 F in each case showed that these 2 quantities stood in the same proportion for each refrigerant, thus establishing an ***empirical*** relationship between the 2 quantities. The heat required to raise the weak liquor from 95 to 232 F could be calculated for Refrigerant 133a, Refrigerant 31 and Refrigerant 134. This was done and the ratio to the value for Refrigerant 21 for this quantity was assumed, on the basis just stated, to be the same as for the total heat requirements.

The foregoing empirical relationship signifies physically that the main heat losses are due to incomplete heat exchange. (Only 36.7 percent of the heat available in the weak liquor from the generator is recovered, and there is no heat exchange between refrigerant entering and leaving the evaporator.)

New solubility data[4], Fig. 1, and published thermodynamic data[5] have been used in preparing Table 1. Unpublished duPont data have also been used where needed and available on Refrigerants 133a, 31, 124a and 134. Where data were unavailable they have been estimated approximately. Such values are indicated by parentheses. The calculation for Refrigerant 30 is based on data from Reference 1.

In Table 1 the arrangement of the fluoroalkane refrigerants is that of decreasing boiling point from left to right. The only nonfluorinated compound, Refrigerant 30 (methylene chloride), is not placed in order of boiling point but is in the last column. It properly belongs by itself because of its low stability as compared to the fluorinated refrigerants. Stability is discussed later.

The end result of the calculation is expressed in item (41), the coefficients of performance (COP). Rearranging on the basis of the order of COP,

REFRIGERANT		COP
21	($CHCl_2F$)	0.375
22	($CHClF_2$)	0.268
31	(CH_2ClF)	0.210
133a	($CH_2Cl \cdot CF_3$)	0.205
30	(CH_2Cl_2)	0.198
124a	($CHF \cdot CClF_2$)	0.195
134	($CHF_2 \cdot CHF_2$)	0.143

Refrigerant 21 thus is best on the basis of its thermodynamic and physical properties. Unfortunately, its chemical stability is not good. Refrigerant 22, in second place in COP, is excellent in chemical stability and is the preferred fluoroalkane refrigerant. The solubility of Refrigerant 22 in dimethyl ether of tetraethylene glycol over a range of temperature, and pressures is given in Fig. 1, which also may be used for calculations at other cycle conditions.

It is to be emphasized that the COP of 0.268 for Refrigerant 22 is for the specific equipment and cycle of Zellhoefer in Reference 3. In this cycle the efficiency of the absorber-generator system is 75 percent as compared to theoretical solubilities and the recovery of heat from the liquor leaving the generator is less than 37 percent. Any considerable improvement in these figures by more effective spraying or better heat transfer would lead to considerable improvement in the COP.

The least desirable feature of Refrigerant 22 as an absorption refrigerant is its relatively high pressure, which leads to power requirements more than 3 times as

TABLE 1. . . .COMPARISON OF THE PERFORMANCES OF ABSORPTION REFRIGERANTS

REFRIGERANT		21	133a	31	124a	134	22	30
FORMULA	STEPS	$CHCl_2F$	$CH_2Cl\cdot CF_3$	CH_2ClF	$CHF_2\cdot CClF_2$	$CHF_2\cdot CHF_2$	$CHClF_2$	CH_2Cl_2
Molecular Weight		102.9	118.5	68.5	136.5	102.	86.5	84.9
Boiling Point, C		+8.9	+6.1	−9.1	−10.2	−23.	−40.8	+40.
Boiling Point, F		+48.1	+43	+16	+14	−9.4	−41.4	+104
Latent heat of vaporization at 40 F								
Btu per lb	(1)	105.2	80.2	136.6	69.4	87.8	87.4	150.5
Btu per lb mole	(2)	10,820.	(9,500)	9,360	9,470	8,950	7,560	12,780
Solubility in E-181								
Absorber								
Operating temperature, F		95	90	95	95	95	95	95
Operating pressure, psia		12.3	13.8	24.2	26.1	41	83.7	3.3
Mole fraction refrigerant	(3)	0.680	0.630	0.601	0.590	0.613	0.748	0.523
Weight percent refrigerant	(4)	49.6	48.2	31.7	46.9	42.1	53.6	29.5
Moles refrigerant per mole E-181	(5)	2.125	1.703	1.506	1.439	1.584	2.97	1.093
Generator								
Operating temperature, F		232.	230.	232.	232	232	232.	232.
Operating pressure, psia		42.9	46.3	72.3	81.4	115.	224.6	14.7
Mole fraction refrigerant	(6)	0.414	0.374	0.366	0.308	0.413	0.516	0.317
Weight percent refrigerant	(7)	24.7	24.2	15.1	21.5	24.4	29.3	15.0
Moles refrigerant per mole E-181	(8)	0.706	0.597	0.577	0.445	0.704	1.07	0.464
Refrig. evolved in generator: (5) − (8)	(9)	1.419	1.105	0.929	0.994	0.880	1.90	0.632
Vapor pressure of refrig., psia at 104 F	(10)	42.9	46.3	72.3	81.4	115.	224.6	14.7
Vapor pressure of refrig., psia at 40 F	(11)	12.3	13.8	24.2	26.1	41.	83.7	3.3
Pressure drop to absorber: (10) − (11)	(12)	30.6	32.5	48.1	55.3	74.	140.9	11.4
(9) × (2)	(13)	15,360.	10,500.	8,690.	9,413.	7,880.	14,364.	8,075.
Amounts Circulated, lb/min/ton Refrig.								
Refrigerant circulated to evaporator	(14)	2.250	3.324	1.739	3.872	3.086	2.995	1.517
Corresponding amount E-181, based on (4) and (7):	(15)	3.425	5.643	5.025	6.343	7.641	4.051	6.284
Actual E-181 based on 75% eff. of absorber-generator system (15)/0.75	(16)	4.567	7.524	6.701	8.457	10.187	5.401	8.379
Refrigerant in weak liquor based on (16)	(17)	1.494	2.397	1.192	2.316	3.269	2.238	1.479
Total amount refrigerant (14) + (17)	(18)	3.744	5.721	2.931	6.188	6.375	5.233	2.996
Total to generator (16) + (18)	(19)	8.311	13.245	9.632	14.645	16.542	10.634	11.375
Specific gravity liquid refrigerant at 60 F	(20)	1.379	(1.365)	(1.226)	1.299	(1.192)	1.213	1.336

Amounts Refrigerant gpm/ton refrigeration based on (18) and (20)	(21)	0.325	0.502	0.286	0.571	0.641	0.517	0.269
Amount E-181 gpm/ton refrig. based on (16) and sp. gr. 1.013	(22)	0.540	0.890	0.793	1.000	1.205	0.6[illegible]	0.991
Total gpm/ton refrigeration (21) + (22)	(23)	0.8656	1.392	1.079	1.571	1.846	1.1[illegible]6	1.260
Power Requirements								
Head + Frictional Losses psia: (12) + 50	(24)	80.6	82.5	98.1	105.3	124	190.9	77.4
Power per Ton Refrigeration								
Pump hp 0.000584 × (23) × (24)	(25)	0.0408	0.0671	0.0618	0.0966	0.134	0.129	0.0452
Estimated actual hp: (25)/0.462[a]	(26)	0.088	0.145	0.134	0.209	0.289	0.279	0.0978
Btu/min theoretically equivalent to (26): 42.41 × (26)	(27)	3.73	6.15	5.68	8.86	12.26	11.8	4.15
Btu/min "Practically" equivalent on a cost basis[b] to (26):(27) × 5.6	(28)	20.9	34.4	31.8	49.6	68.6	66.1	23.2
Heat Requirements								
Specific heat of liquid refrigerant at 77 F Btu/(lb) F	(29)	0.255	0.313	0.337	0.292	0.359	0.298	0.287
Heat Capacity in Btu/(min) (ton refrigeration)								
Total refrigerant (29) × (18)	(30)	0.955	1.791	0.988	1.807	2.289	1.559	0.860
Total E-181: 0.447 × (16) = (31)	(31)	2.041	3.363	2.995	3.780	4.554	2.414	3.745
(30) + (31) = (32)	(32)	2.996	5.154	3.983	5.587	6.842	3.973	4.605
Ratio to value for Refrigerant 21: (32)/2.996	(33)	1.000	1.720	1.329	1.865	2.284	1.326	1.537
Overall Comparison with Refrigerant 21								
Heat required[c]								
By refrigerant 137 × (17) × (29)	(34)	52.2[d]	102.8	55.0	92.6	162.1	91.4	112.0
By absorbent 137 × (16) × 0.447	(35)	279.7	491.7	437.9	517.9	665.7	330.7	513.1
Total: (34) + (35) = (36)	(36)	331.9	594.5	492.9	610.5	827.8	422.1	625.0
Ratio[e] of values to Refrigerant 21 value: (36)/331.9	(37)	1.00	1.79	1.49	1.839	2.49	1.27	1.88
Percentage excess Refrigerant 21 value								
Due to increased heat requirements (37 − 1.00) × 100	(38)	0	79	49	84	149	27	88
Due to increased power requirements 100 [Step (28) − 20.9)/(333 + 20.9]	(39)	0	3.8	3.1	8.1	13.5	12.8	0.7
Total (38) + (39)	(40)	0	83	52	92	163	40	89
Coefficient of Performance (For R 21: COP = 200/533 = 0.375) COP = 0.375/[(100 + (40))/100]	(41)	0.375	0.205	0.247	0.195	0.143	0.268	0.198

[a] Pump efficiency of 0.66 times motor efficiency of 0.70. [b] Cost per kwhr 2c and cost per therm. 10.5c. [c] To raise weak liquor from 95 to 232 F, Btu per (min) (ton). [d] Value from the thermodynamic tables instead of approximate calculation indicated by the equation. [e] These ratios are also the ratios of the respective total heat requirements within our accuracy.

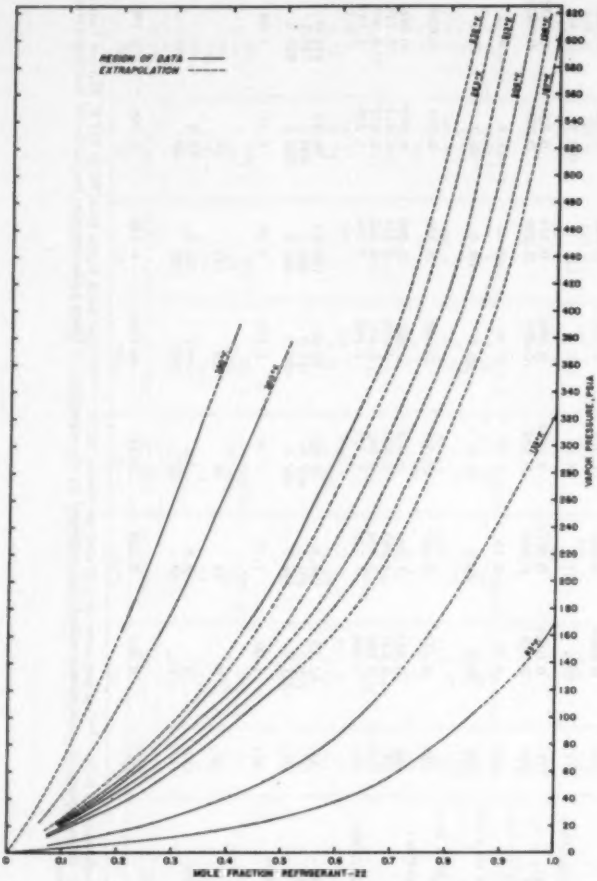

FIG. 1....RECENTLY DEVELOPED SOLUBILITY DATA FROM DUPONT'S JACKSON LABORATORY

great as for Refrigerant 21 (see item (26), Table 1). The additional power, however, need not be wasted, since with correct design it can, in considerable part, be utilized to decrease film coefficients in the heat exchanger. Stating this in another way: It would be necessary to put power in to achieve good heat transfer, were it not that this power has already been made available in pumping from the low side to the high side.

The factors having the largest effect on performance are the difference in solubility of the refrigerant in the absorber and in the generator and the latent heat of

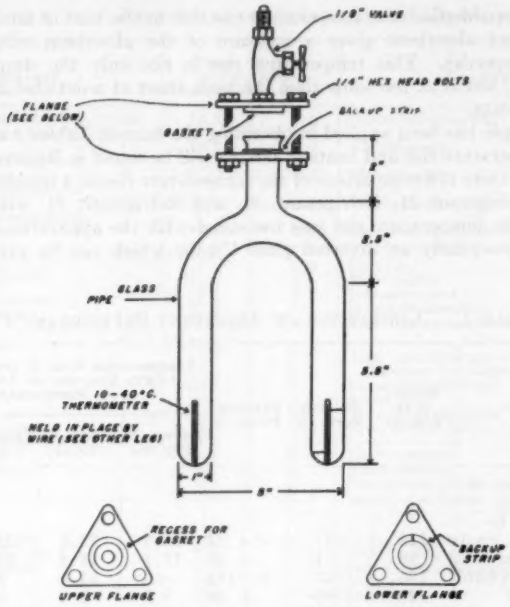

FIG. 2. . . .NITROGEN-PRESSURIZED U-TUBE USED TO MEASURE TEMPERATURE RISE

the refrigerant. The best solubility characteristics are exhibited by Refrigerant 22. This may be seen from Item 9, Table 1, which gives the moles of refrigerant evolved in the generator per mole of dimethyl ether of tetraethylene glycol circulated. Despite its relatively poor solubility characteristics, Refrigerant 31 is in third place in coefficient of performance because its high latent heat per pound entails circulation of comparatively little refrigerant. The latent heat of Refrigerant 22 is relatively low, so that its COP falls behind that of Refrigerant 21 which has a better latent heat, and solubility characteristics only moderately below those of Refrigerant 22.

As just discussed, a major problem in absorption refrigeration is that of keeping the quantity of absorbing liquid small. To do this, it is necessary to have a large difference in solubility of the refrigerant in the absorbent between the absorber conditions and the generator conditions. This entails a large change in solubility with temperature. In order to have a large decrease in the solubility of the refrigerant with increasing temperature, there must be a large evolution of heat† on mixing the liquid refrigerant and absorbent. Whence it results that to a good

† Statements may be found to the effect that a large heat of mixing is undesirable, on the basis that this heat of mixing must be removed in the absorber and introduced in the generator. It is true that the heat of mixing must be introduced and removed as stated, but that is the price that has to be paid for the required large change in solubility between absorber and generator conditions.

degree of approximation, the temperature rise due to the heat of mixing the liquid refrigerant and absorbent gives a measure of the absorbent-refrigerant pair's solubility properties. This temperature rise is not only the simplest relevant measurement, but is at the same time the best, short of a detailed analysis based on complete data.

This principle has been applied in obtaining the data in Tables 2 and 3. Other data on temperature rise and heats of mixing will be found in Reference 6.

In Table 2 there is a comparison of the temperature rise of 4 refrigerants, Refrigerant 22, Refrigerant 21, Refrigerant 30, and Refrigerant 31, with several absorbents. The temperature rise was measured with the apparatus shown in Fig. 2, which is essentially an inverted glass U-tube which can be pressurized with

TABLE 2. . . .COMPARISON OF ABSORBENT-REFRIGERANT PAIRS

ABSORBENT	MOLECULAR WEIGHT	BOILING POINT, F	FREEZING POINT, F	TEMPERATURE RISE, F, ON MIXING EQUAL LIQUID VOLUMES OF ABSORBENT AND REFRIGERANT			
				REFRIG. 22 ($CHClF_2$)	REFRIG. 21 ($CHCl_2F$)	REFRIG. 30 (CH_2Cl_2)	REFRIG. 31 (CH_2ClF)
Dimethyl ether of tetraethylene glycol (DME-TEG)	222	527	< − 22	24.3	23.4	22.1	14.4
Cyclohexanone	98	311	− 49	17.1	15.3	10.8	7.6
Tributyl phosphate	266	552	< −112	——	14.4	8.1	——
Acetophenone	120	396	+ 69	9.9	9.0	5.4	4.5
Isophorone	138	419	——	12.6	9.0	7.2	0.9

nitrogen. It permits ready mixing of relatively high-pressure refrigerants at room temperature.

The procedure is to introduce the absorbent into one leg and the chilled refrigerant into the other leg. The apparatus is then pressurized (behind a safety shield) with nitrogen to above the saturation pressure of the refrigerant at room temperature. This prevents the refrigerant from distilling over into the absorbent. The absorbent and refrigerant are brought to room temperature and are rapidly mixed by tilting the tube and swirling.

The data of Table 2 show the same order of temperature rise with the several absorbents, with Refrigerant 22 highest, Refrigerant 21 next, then Refrigerant 30 (methylene chloride) and finally Refrigerant 31. Accordingly, if for a series of absorbent candidates, the temperature rise with one of these refrigerants is known, one can reach conclusions as to the relative rise with other refrigerants. Thus, it suffices to measure the temperature rise with Refrigerant 30, which is particularly convenient since pressure apparatus is not required in handling it. Extensive measurements with Refrigerant 30 are reported in Table 3. The figures of Table 3 are comparable among themselves, but are not always identical with those of Table 2, since different apparatus was used (consisting simply of graduated cylinders and thermometers used as stirring rods for the CH_2Cl_2 measurements). The heat transfer to the glass and surrounding air did not have to be corrected for, since interest is only in relative values of the temperature rise.

TABLE 3. . . .COMPARISON OF ABSORBENT-CH_2Cl_2 PAIRS

ABSORBENT	TEMP. RISE ON MIXING EQUAL LIQUID VOLUMES, F		ABSORBENT
1-Ethyl-3-methyl-phospholine-1-oxide	25.9	5.6	N-Methylformamide
Tetraethylene glycol dimethyl ether (DME-TEG)	22.1	5.4	Acetophenone
DME-TEG/Tetramethylurea (75/25 by volume)	21.2	5.4	Methyl sulfite
DME-TEG/Tetramethylurea (50/50 by volume)	20.9	5.4	Triethyl amine
DME-TEG/Tetramethylurea (25/75 by volume)	19.4	5.2	Experimental Polyketone C
Tetramethylurea	18.4		
N,N-Dimethylacetamide	17.6	5.0	Pyridine
Dimethyl sulfoxide	12.6	3.6	Ethyl sulfide
N,N-Dimethylformamide	12.6	3.6	Nitropropane-1
2,6-Lutidine, N-oxide	12.6	3.6	Ethylene phosphite
		3.6	Experimental polyketone D
Diisobutyl phosphite	12.6		
Tetrahydrofuran	11.2		Acetylated monoglycerides
Diethylcyanamide	11.0	2.7	(Eastman Type-940)
Cyclohexanone	10.8	2.7	Ethylaulfonylethanol
Adiponitrile	10.8	1.8	Acetic acid
		1.8	Triisobutyl phosphite
Diisopropylcyanamide	9.9	1.8	Experimental polyketone E
Experimental Polyketone A	9.9		
Tributyl phosphate	8.1	0	1,4-Butanediol
Dimethylcyanamide	7.7	0	Butyl disulfide
Isophorone	7.2	0	Poly(ethylenimine)
		−1.8	Dimethyl sulfolane
Polyethylene glycol 200	7.2	−2.7	Dow Corning Silicone #200
Experimental Polyketone B	7.2	−2.7	Dow Corning Silicone #1107
Tetramethylene sulfone	6.7	−2.7	Dow Corning Silicone #1108
Ethyl ether	6.3	−3.6	Aniline
Acetone	6.3	−3.6	Methanol
		−7.2	Anhydrous ethanol
		−9.4	95% Ethanol

TABLE 4. . . .COMPARISON OF STABILITIES OF REFRIGERANTS 21 AND 30

(System: 2 cc Refrigerant + 2 cc Naphthenic Oil or DME-TEG + metals. Temperature: 250 F)

REFRIGERANT	DURATION TEST, DAYS	REFRIG. DECOMPOSED, %	PENETRATION, IN. PER MO × 10^6			COMMENTS
			ALUMINUM	COPPER	STEEL	
A. Tests with naphthenic oil						
21	1	0.003	420	2,000[b]	500[b]	
30	1	4.0	130,000	1,800[b]	1,500[b]	general decomposition in one day
30	1	[a]	[a]	[a]	[a]	carbonized in one day.
B. Tests with DME-TEG absorbent						
21	8	0.34	100	260[b]	140[b]	
30	8	10.2	40,000	160[b]	220	average of two tubes.

[a] = no data. [b] = gain in weight.
Note: Refrigerants: highest purity, center-cut material.

The data of Tables 2 and 3 show the dimethyl ether of tetraethylene glycol (DME-TEG) to have most desirable solubility properties. This is confirmed by the solubility data given in Reference 4 cited previously. It also has a desirably low vapor pressure and viscosity and is reported to be low in toxicity (Reference 7). Furthermore, it has good chemical stability. The only substance in Table 3

TABLE 5. . . .COMPARISON OF STABILITIES OF REFRIGERANTS 12, 21 AND 22 WITH NAPHTHENIC OIL PRESENT

(System: 2 cc Refrigerant + 2 cc Naphthenic Oil + Copper + Mild Steel. Temperature: 250 F. Duration: 60 days.)

REFRIGERANT	REFRIGERANT DECOMPOSED, %	PENETRATION, IN. PER MO × 10^6		COMMENTS (AVG OF)
		COPPER	STEEL	
22	0.001	2	7	3 tubes
12	0.032	1	13	2 tubes
21	0.21	4	13	2 tubes
21[a]	[b]	6	19	2 tubes

[a] Highest purity, center-cut material; other refrigerants are commercial grade (99.8+ %).
[b] no data.

giving a greater temperature rise, 1-ethyl-3-methyl-phospholine-1-oxide, failed to show good stability with Refrigerant 22.

Other conclusions from the data of Tables 2 and 3 are:

1. In addition to polyethers, other desirable configurations for high solubility are carbonyl, sulfoxide, N,N dialkyl amide, P-oxide, N-oxide and nitrile.

2. Mixtures of the dimethyl ether of tetraethylene glycol with tetramethyl urea showed essentially additive properties.

The chemical stability of a system depends not merely on the stability of the individual components, but also on their mutual effects, which may be large. Therefore, all the components should be included simultaneously in stability tests. In view of the results of the study of absorbents, above, the dimethyl ether

TABLE 6. . . .COMPARISON OF STABILITIES OF REFRIGERANTS 21, 22 AND 31 WITH DME-TEG ABSORBENT PRESENT

(System: 2 cc Refrigerant + 2 cc DME-TEG + Copper + Mild Steel. Temperature: 250 F. Duration: 60 Days.)

REFRIGERANT	PENETRATION (IN. PER MO × 10^6)		COMMENTS
	COPPER	STEEL	
22	3	6	
31	4	42	
21	470	1200	Strong corrosion
21[a]	490	1100	Strong corrosion

Note: Each entry is the average of two tests.
[a] Highest purity, center-cut material; other refrigerants are commercial grade (99.8+ %).

of tetraethylene glycol was selected as the absorbent to be used in the stability tests.

Refrigerant, absorbent, and coupons of metals of construction were loaded into glass pressure tubes (Reference 8) and sealed by fusing the glass with the tube evacuated to less than 0.03 mm. of mercury pressure, while the contents of the tube were frozen at liquid nitrogen temperature. In this way the glass at the top was fused without decomposing any refrigerant or other material frozen in the bottom of the tube. The tubes were stored at elevated temperature during periods up to 90 days. Inspections were made at regular intervals. At the end of the storage period, the tubes were opened and analyzed for chloride ion formation, from which the amount of refrigerant decomposed could be calculated. The penetrations of the metals were also determined. These quantitative measures of decomposition usually correlated well with each other and with the extent of deterioration indicated by change in appearance.

Several groups of data were obtained. One compared Refrigerant 30 with the fluorinated refrigerants. Another compared the stabilities of the 3 fluoroalkanes which were most attractive from the thermodynamic point of view, namely, Refrigerants 21, 22 and 31. A third group gave more extensive information on the most stable of these, Refrigerant 22. Also included were tests in which a naphthenic-type refrigeration grade oil was substituted for the DME–TEG absorbent. These latter represent standard compressor system stability tests and can be compared with broad experience with many refrigerants.

The conclusions reached are as follows:

1. The only nonfluorinated refrigerant studied, Refrigerant 30, is far inferior in stability to Refrigerant 21, as may be seen in Table 4. This is true both with DME-TEG absorbent and with refrigeration-grade oil.

2. However, Refrigerant 21 itself shows relatively poor stability, decomposition being appreciable within a few days at 250 F in all Refrigerant 21 systems tested. In particular, in Table 5, Refrigerant 21 is much inferior to Refrigerants 12 and 22 with naphthenic oil, copper and mild steel at 250 F. Similarly, Refrigerant 21 is shown, in Table 6, to be much inferior to Refrigerants 31 and 22 with DME-TEG absorbent, copper and mild steel. That the DME-TEG is not responsible is shown in Tables 7 and 9. While Refrigerant 21 alone is not much worse than DME-TEG, Table 7, the low stability of the mixture must be attributed to Refrigerant 21, in view of the data of Table 4 showing low stability for the Refrigerant 21-DME-TEG mixture and Table 8 showing that replacement of Refrigerant 21 in this mixture with Refrigerant 22 leads to high stability.

3. Refrigerant 31 is much better in stability than Refrigerant 21, but appreciably inferior to Refrigerant 22. This is shown by the data of Table 6.

4. Refrigerant 22 shows excellent chemical stability with DME-TEG absorbent with aluminum, copper and steel at temperatures up to at least 350 F, as may be seen in Table 8.

TABLE 7. . . .STABILITY TESTS ON SINGLE LIQUIDS

(System: 2 cc Liquid + Copper + Mild Steel. Temperature: 250 F. Duration: 60 Days.)

LIQUID	PENETRATION, IN. PER MO × 10^3		COMMENTS
	COPPER	STEEL	
Refrigerant 21	5	9	Slight corrosion
DME-TEG	3	5	Metals bright

Note: Results are average of two tests.

5. In the Refrigerant 22 + DME-TEG systems (Table 8), least decomposition occurs in the absence of mild steel. However, the amount of decomposition is small even when steel is present.

6. In the Refrigerant 22 + DME-TEG systems (Table 8) no gain in stability is indicated for a reduction in moisture in the absorbent from 460 ppm. (commercial

TABLE 8. . . .EFFECT OF TEMPERATURE ON STABILITY OF REFRIGERANT 22—DME-TEG SYSTEM WITH VARIOUS METAL COMBINATIONS

(System: 2 cc Refrigerant 22 + 2 cc DME-TEG + metals.)

TEMP, F	DME-TEG[a]	REFRIG. DECOMPOSED, %	PENETRATION, IN. PER MO × 10^3			DURATION OF TEST, DAYS	NUMBER OF TESTS
			ALUMINUM	COPPER	STEEL		
250	commercial	0.001	7	2	2	61	3
300	commercial	0.12	9	5	12	90	3
300	anhydrous	0.19	11	2	16	90	1
350	commercial	0.13	15	1	13	90	3
350	anhydrous	0.13	15	2	17	90	2
300	commercial	0.15	—	2	17	90	3
300	anhydrous	0.33	—	1	37	90	2
350	commercial	0.030	—	2	9	90	3
350	anhydrous	0.046	—	4	36	90	2
300	commercial	0.003	4	1	—	90	3
300	anhydrous	0.002	6	2	—	90	2
350	commercial	0.003	2	2	—	90	3
350	anhydrous	0.002	1	2	—	90	2

[a] Commercial DME-TEG: 460 ppm moisture; 2.3 mg *active hydrogen* per 100 ml (by a lithium aluminum hydride method). *Anhydrous* DME-TEG: 35 ppm moisture; 1.75 mg *active hydrogen* per 100 ml (by a lithium aluminum hydride method).

grade DME-TEG) to 35 ppm. ("anhydrous" DME-TEG). What difference there is, is in favor of the commercial grade.

7. In the Refrigerant 22 + DME-TEG systems, the effect of raising the temperature from 250 to 300 F was to increase the attack, as was to be expected. However, unexpectedly, this was not the case on raising the temperature from 300 to 350 F.

8. In the absence of refrigerant, DME-TEG alone, commercial or "anhydrous," does not attack aluminum, copper and mild steel appreciably (Table 9) at temperatures up to 350 F.

9. In the case of Refrigerant 22 + DME-TEG, mild steel has only a small adverse effect on stability at 300 and 350 F, as pointed out under 5 above. However, in the cases of Refrigerant 31 and particularly Refrigerant 21 + DME-TEG, the adverse effect of steel is pronounced even at 250 F, as may be seen in Table 10. In the absence of steel, other metals, namely aluminum, copper and 18-8 stainless steel, were little attacked and Monel was affected only slightly more.

It then appears that only Refrigerant 22 can be expected to provide long life in the systems considered, when mild steel is present. The Refrigerant 22 systems

show good stability under the relatively severe conditions represented by 90 days at 350 F.

Of the halogenated absorption refrigerants, Refrigerant 21 would be the most attractive, were it not for its relatively poor chemical stability. Refrigerant 22, with excellent chemical stability, a necessity for long useful life of practical units, as well as good thermodynamic properties is the preferred halogenated refrigerant. After an extensive study, the dimethyl ether of tetraethylene glycol (DME-TEG) remains the preferred absorbent. The basis for the conclusion that the Refrigerant 22-DME-TEG system is attractive has been developed fully above. How does this fluoroalkane refrigerant system compare with other absorption systems?

In attempting to answer this question many factors must be considered. Table 11 cross-compares the principal systems with regard to important characteristics.

TABLE 9. . . . EFFECT OF DME-TEG ON VARIOUS METAL COMBINATIONS AT VARIOUS TEMPERATURES

(*System: 2 cc DME-TEG + Metals.*)

TEMP, F	DME-TEG[a]	PENETRATION, IN. PER MO × 10^6			DURATION OF TEST, DAYS	NUMBER OF TESTS
		ALUMINUM	COPPER	STEEL		
250	commercial	none	3	5	60	2
300	commercial	none	1	1	90	1
350	commercial	none	2	2	90	1
300	anhydrous	3	1	3	90	1
350	anhydrous	2	0	2	90	1
300	commercial	4	2	none	90	1
350	commercial	3	1	none	90	1

[a] Commercial DME-TEG: 460 ppm moisture; 2.3 mg *active hydrogen* per 100 ml (by a lithium aluminum hydride method). *Anhydrous* DME-TEG: 35 ppm moisture; 1.75 mg *active hydrogen* per 100 ml (by a lithium aluminum hydride method).

If the units are to be air cooled, the lithium bromide-water system is excluded. If refrigeration, not air conditioning, is the objective, the lithium bromide-water system is excluded. The water-ammonia system is dangerous because of the toxicity and flammability of ammonia and the explosive character of its mixtures with air. In all of these respects the Refrigerant 22 system is superior. However, it must be borne in mind that DME-TEG, although not easily ignited, in view of its flashpoint of 285 F, will burn if heated hot enough. Its presence means one does not have the complete freedom from flammability hazards which Refrigerant 22 itself possesses. Also it must be remembered that while the Refrigerant 22 + DME-TEG system can be used with an air-cooled condenser or a low temperature evaporator, the coefficient of performance will decrease as the condenser temperature increases or the evaporator temperature decreases, other things remaining unchanged.

In the matter of materials of construction, the Refrigerant 22 system offers more flexibility than the others. Iron, copper, aluminum and many alloys can be used. Lithium bromide solution is quite corrosive, requiring special alloys. Ammonia is incompatible with copper.

The ideal absorbent is completely nonvolatile and readily separated from the refrigerant by distillation. Lithium bromide, while nonvolatile, tends to foam

TABLE 10. . . .EFFECT OF VARIOUS METALS ON THE STABILITIES OF REFRIGERANTS WITH DME-TEG ABSORBENT

(System: 2 cc Refrigerant + 2 cc DME-TEG + Metals. Temperature: 250 F. Duration: 60 Days.)

REFRIGERANT	REFRIG. DECOMPOSED, %	PENETRATION, IN. PER MO × 10⁶					COMMENTS
		ALUMINUM	COPPER	STEEL	18-8 STAINLESS STEEL	MONEL	
A. COMBINATIONS EXCLUDING MILD STEEL							
31	0.08	1[b]	0	—	—	—	
31	0.10	7	3	—	1[b]	—	
21	[a]	—	2	—	3	—	
21	[a]	2[b]	—	—	6	9	Monel tarnished
B. COMBINATIONS INCLUDING MILD STEEL							
22	[a]	—	3	6	—	—	
31	0.43	—	4	42	—	—	
21	[a]	—	490	1,100	—	—	Strong Corrosion
21	[a]	6,400	—	510	—	—	Aluminum partly eaten away
21	[a]	—	—	640	570	—	Metals badly corroded
21	[a]	—	—	1,200	—	400	Metals badly corroded

[a] = no data. [b] = gain in weight.

Note: Refrigerants 21 and 31 are highest purity center-cut material; Refrigerant 22 is commercial grade (99.8+ %).

TABLE 11. . . .RELATIVE MERITS OF THREE ABSORPTION SYSTEMS

CHARACTERISTIC	REFRIGERANT-ABSORBENT PAIR		
	WATER–LITHIUM BROMIDE SOLUTION	AMMONIA–WATER	REFRIG. 22–DME-TEG
Can be air cooled	no	yes	yes
Can be used for refrigeration	no	yes	yes
Refrigerant nonflammable	yes	no	yes
Refrigerant nontoxic	yes	no	yes
Absorbent nonflammable	yes	yes	no[a]
Absorbent nontoxic	yes	yes	yes
Components noncorrosive	no	yes[b]	yes
Nonvolatility of absorbent	excellent but tends to foam over	poor	good
Latent heat of refrigerant	excellent	good	fair
Operating pressure	extremely low	high	high

[a] Flash point of absorbent is high, 285 F.
[b] Corrosive to copper so that copper and copper alloys must be excluded.

over into the parts of the system from which it is desired to exclude it. In the ammonia system the absorbent, water, is far too volatile, requiring additional *rectifiers* and *analyzers* to remove water from the refrigerant. DME-TEG, boiling at 528 F, has good properties in this respect.

In regard to latent heat of vaporization, water as refrigerant is outstanding and ammonia quite good. A high latent heat means that relatively little refrigerant is required and that comparatively small amounts of liquids need to be circulated in the system. As a result, heat transfer surface requirements are low. A direct cross comparison, such as just made on the fluoroalkanes, of Refrigerant 22 with water or ammonia has not been possible because of the radically different absorbents and quite different apparatus involved.

There is, however, no doubt that the high latent heats of water and, to a lesser extent, ammonia will be an important factor in keeping down heat transfer surface requirements. How much heat transfer must be achieved will depend on what coefficient of performance is required.

Both the ammonia and Refrigerant 22 systems operate at relatively high pressures. While this minimizes the noncondensables problem, that bane of the extremely low-pressure water lithium-bromide system, it means relatively high pumping power requirements. However, as pointed out earlier in this paper, with proper design this power need not be wasted but can, in considerable part, be utilized to improve heat transfer and thereby coefficient of performance as discussed.

While all of the systems considered in this study have certain deficiencies it is clear that the Refrigerant 22-dimethyl ether of tetraethylene glycol system has enough advantages to merit careful consideration for all absorption refrigeration applications.

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DISCUSSION

H. O. SPAUSCHUS, Louisville, Ky.: The screening method used by the author in selecting solvents depended on measuring the heat of mixing. Was the effect of final composition and temperature taken into consideration? It has been established by Bambach [Kältetechnik, 7, 187 (1955)] that the heat of mixing of Refrigerant 12 and mineral oil

is strongly temperature and concentration dependent making it exceedingly difficult to obtain accurate colorimetric data.

Secondly, was the order of solubility studied on a mole fraction basis as well as on the basis of mixing equal volumes of the two components?

LEONID OURUSOFF, Washington, D. C.: On one of the illustrations the coefficient of performance for Refrigerant 21 was shown as roughly 0.37 and for Refrigerant 22 as roughly 0.27. Where on that comparison would ammonia-water be placed?

Would the Refrigerant 22 be more in the same direction as ammonia but perhaps not quite as high? Would ammonia still be between the two?

J. C. LEE, Des Plaines, Ill.: Is it correct that the COP values calculated are based on the assumption that there is no heat exchanger in the cycle?

Another comment is in connection with basing the absorber temperature at 95 F when for air cooling it is necessary to go at least to 140 F. What would the generator temperature be with an absorber temperature of 140 F?

On looking at the vapor pressure characteristics on the graph, it appeared that the temperature may be well over 300 F. Now, the question comes up whether the absorbent will have enough vapor pressure at that temperature to necessitate the use of a bubble tower.

AUTHOR'S CLOSURE: The answer to both inquiries of Mr. Spauschus is that this was merely a screening procedure. What was done was to obtain actual solubility data on the systems which had shown an attractive temperature rise.

Concerning the first question of Mr. Ourusoff it proved impossible to make a fair comparison because of differences in apparatus and operating conditions. The answer to his question is believed to be that, if the same amount of heat transfer surface is used, results would be better with the ammonia-water pair. An important factor is the relatively high latent heat of vaporization of ammonia, so that smaller volumes are required. There are, however, various other factors of importance, such as utilizing the pumping energy, the design of the apparatus, as well as others. Where the ammonia-water pair would fall as compared to Refrigerants 21 and 22, would depend on the apparatus, which would be quite different in the two cases. The author is not prepared to say whether ammonia would fall between the two.

With regard to Mr. Lee's inquiry about a heat exchanger in the cycle, in Zellhoefer's work which is cited in the paper, there was a heat exchanger for the weak liquor coming from the generator and the liquid coming from the absorber, but the efficiency was only about 37 percent. The "efficiency" means that the total amount of sensible heat available which could have been utilized was recovered only to the extent of 37 percent. What could have been recovered usefully was recovered to that extent.

Mr. Lee also asked what the generator temperature would be with an absorber temperature of 140 F. That was not calculated. This study was for a water-cooled condenser. Zellhoefer in his paper used water cooling and what has been done here is to go through the substitution described, of one refrigerant for another, with suitable adjustments. The answer to discusser's question is not available, but could be estimated from the graph of the solubility of Refrigerant 22. The exact points wanted may not be on the chart, but by interpolating an approximate measure of the solubility relationships can be obtained.

In reply to Mr. Lee's last question, as the temperature rises the vapor pressure certainly increases. It was not necessary at 232 F to use a rectifier. Just at what point it would become desirable depends on economics, and has not been determined. So there is no direct answer on that, but the vapor pressure of the absorbent is known. There are published data cited in the references in the paper. Thus the information needed to make this estimate is available.

1692

No. 1692

ESTIMATING WATER CONTENT OF CERTAIN DRIED BUT UNCHARGED HERMETIC REFRIGERATING COMPRESSORS

By H. M. Elsey*, Oakmont, Pa., J. B. Kelley**, and R. B. Sharpe†, Columbus, Ohio

THE TEST equipment used in the centrifuge-cold trap investigations here reported is illustrated, schematically in Fig. 1; details in Fig. 2.

A dried and sealed compressor, A, was placed in a 300 F oven and evacuated through a train which included a glass trap, C, cooled to approximately −105 F by immersion in a trichloroethylene-dry ice mixture in D. At the end of 3 hours the valve, B, was closed on the compressor, the trap was removed from the cold bath, warmed to room temperature and whirled in a centrifuge for a minute or less. This step forced the liquid condensate into the graduated capillary tube which formed the bottom of the trap.

The liquid frequently separated into two layers. The upper, or less dense layer, consisted of low boiling petroleum hydrocarbons from the solvent which had been used to lubricate the compressor during a pump test. The lower and denser layer was water, the weight of which could be estimated directly in milligrams from the scale reading.

This method was developed for use with a type of compressor unit in which the stator and its housing was pressed tightly into the steel shell of the unit. The water-immersion leak-test, which immediately followed the oven dehydration, rapidly cooled the motor so that by the time a compressor was placed in the oven for a moisture check, the cellulose insulation was practically at room temperature. Fortunately, the same excellent heat-conducting path, which existed between the stator iron, the housing and the shell and which incidentally caused the rapid

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† Process Engineer, Refrigerator-Freezer Engineering, Westinghouse Electric Corporation.
‡ Exponent numerals refer to References.
Presented at the Annual Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Lake Placid, N. Y., June 1959.

cooling, also allowed the insulation to be reheated rapidly. Hence, the cellulose was at 290 F, or above, for approximately 2 out of the 3 hr it was in the oven.

Experiments which were made to determine the limits which should be set for the 3-hr cold trap test confirm the conclusions expressed by J. L. Knight[1] who reported that about 12 hr are required to remove by evacuation substantially all of the water from the cellulose insulation of a small hermetic motor when it is heated to 300 F.

Unfortunately, 12 hr is too long a time to wait for moisture results on a refrigerator production line and the authors found it entirely satisfactory to shorten

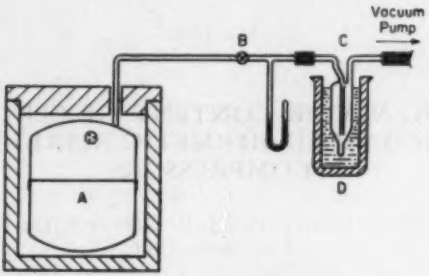

FIG. 1. . . . TEST EQUIPMENT USED IN CENTRIFUGE-COLD TRAP METHOD

the evacuation to 3 hr and modify the limits so as to keep the true water content at a safe level.

Stopping a cold trap test, before substantially all of the water has been removed from the unit under test, is safe however only if the detailed procedures which are used in the shortened test are controlled carefully. Among the essential details are such items as good maintenance of the vacuum pump and the metal and rubber tubings, and careful control of the pumping rate during the early stages of the unit evacuation. Not the least in importance is exact control of the oven temperature.

Recent demand for a faster method of estimating moisture was caused by the adoption of a new internally spring-mounted compressor. Here the good heat-conducting path between the motor and shell is absent, and the cold motor insulation cannot be reheated fast enough, by simple placement of a unit in a hot oven, to arrive at even an approximate estimate of the water content within the 3-hr specified time.

REHEATING STATORS BY GAS PRESSURIZING

Naturally, efforts were first directed toward reheating the stator as fast as possible and leaving the balance of the moisture test procedure unchanged.

There are several ways by which the reheating of a stator can be speeded. For example, the shell can be filled to a pressure of several atmospheres with an inert dry gas such as nitrogen. This will decrease the time required to heat the insulation to 300 F but it will slow down the rate at which the unit can be evacuated to

remove the moisture and transfer it to the cold trap. The mass flow of gas through the cold trap must be low or some of the condensed water will pass through the trap and be lost to the test. This can cause a substantial error in the results.

Use of air instead of nitrogen to improve the reheating rate will completely invalidate the results. For example, a compressor was filled with dry air at a pressure of one atmosphere and held overnight in a 300 F oven. Over 800 mg of

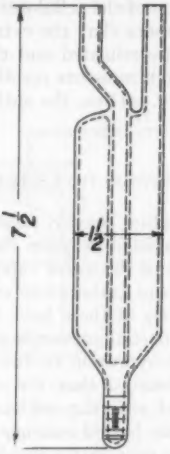

FIG. 2. . . .CLOSE-UP OF CENTRIFUGE COLD TRAP DETAIL

water were found in the cold-trap when it was then evacuated. That this excessive amount of water was formed by the oxidation of oil and other organic matter, rather than by thermal decomposition of the cellulose, is shown by the results of a second experiment. In this, a compressor was filled with one atmosphere of dry oxygen-free nitrogen and left overnight in the same oven. Evacuation through a cold-trap yielded only 100 mg of water.

REHEATING STATORS ELECTRICALLY

Another method of speeding up the reheating of the motor insulation is to carefully apply current to the windings. The copper windings are thermally, as well as electrically, insulated from the massive metal parts of the motor and pump. Thus the wattage which must be applied to the windings to heat the assembly to the desired temperature rapidly will decompose some of the cellulose unless extreme care is used. Overheating to the point of real damage to the insulation will be discovered by the formation of excessive amounts of water which will be a good

indication that the unit should be scrapped. This will be confirmed by the burned odor of the condensate which collects in the trap.

Slight overheating will actually improve the insulating properties of the cellulose and it will not give a burned odor to the condensate although it will increase the water found through thermal decomposition of some of the paper. Thus the interpretation of a result which indicates a wet unit will always be questioned when the windings have been electrically energized during the reheating. The question becomes: Was the drying really unsatisfactory, or has the extra water been formed by the decomposition of the cellulose?

It was the opinion of the authors that the extreme care which must be used when the windings are electrically reheated and the doubt which will always be raised about the validity of a high moisture result make this modification of the trap test definitely unsatisfactory. Thus, the authors continued the search for a good short moisture test method.

DEW-POINT METHOD OF ESTIMATING WATER

Piper[2], on the basis of a literature survey, published a family of curves which showed the relationship which existed between the equilibrium water content of cellulose and the partial pressure of the water vapor in the air around it. Shortly after his paper appeared, the present authors received a full-scale copy of the curves from Mr. Piper. Certain sections of these have been extrapolated to cover the higher temperatures and lower partial pressures, and these modified curves have been extremely useful in these dehydration studies.

For example, it had been observed that the average water content of compressors which were dried at 300 F with the end heads off of the shells, was lower in the winter season than during the humid summer time. A study of the extrapolated Piper curves indicated that more water should be retained, even at 300 F, by cellulose when this was dried while surrounded by humid summer air, than should be retained when it was dried in winter air. The latter often has a low water content.

These predictions were confirmed by a number of experiments which were made in 1953. The results of 3 are cited here.

Experiment 1: 70 g (grams—on a dry basis) of cellulose insulation were dried in a 257 F oven which contained water vapor at a partial pressure of 21 mm of mercury. The *Piper* curves predicted that the dried material should have a residual water content of 190 mg. A cold trap test recovered 170 mg from the sample.

Experiment 2: A second 70-g sample was dried for the same length of time at the higher temperature of 302 F in air which also had a water partial pressure of 21 mm. In this case the curves predicted a residual water content of 80 mg; 70 mg were recovered in the trap test.

Experiment 3: A third 70-g lot of insulation was dried for the same time at the much lower temperature of 212 F, but with the cellulose surrounded by air which was *bone-dry*, that is with a water content approaching zero. The water found in the dried sample by the trap test was only 11 mg which proved that low temperature drying in dry air can be more effective than high temperature drying in wet air. Incidentally, the *Piper* curves were of no help in this last experiment as they do not cover this extremely dry range.

These 3 experiments, together with many others, show the definite retarding effect of air humidity on a drying process where there is an affinity between water and the substance which is being dried. Moreover, they indicate a certain validity for the *Piper* curves.

Results of these three experiments are cited because a study of the *Piper* curves indicated that there should be enough water in the authors' dried compressor units to be estimated by a dew-point method.

Fig. 3 gives a schematic outline of the apparatus used to check the foregoing prediction. A dried compressor, A, was removed from the continuous drying

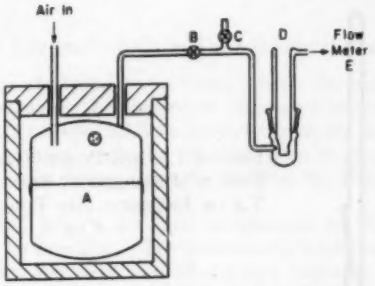

FIG. 3. . . .APPARATUS USED TO CHECK THE PREDICTION THAT WATER IN DRIED COMPRESSOR UNITS MAY BE CHECKED BY DEW-POINT METHOD

oven and fitted with an end-head. This operation was carried out as quickly as possible and the usual immersion leak-test was omitted in order that the compressor would be as hot as possible when it was placed in a foam-resin insulated, close-fitting box.

As soon as the box was closed, the temperature of the shell was followed by means of a thermocouple, which had been attached to the end-head. At the same time, the temperature of the main windings was followed by resistance readings which were taken with a bridge. For a short time the temperature of the shell rose rapidly while that of the windings decreased. After about 40 min the rate at which these 2 temperatures changed, decreased substantially and dew-point readings were taken.

Air, at a very low and constant rate, was then blown into the shell through one of the copper tubes. This air displaced some of the air already in the shell, and this passed out through the second copper tube and valve, B, into the dew-point cell, D, and then out of the system through the sensitive flowmeter, E. Two purposes were served by keeping the flow rate low: *first*, the extent to which the equilibrium was disturbed between the water in the cellulose and that in the air around it was minimized; *second*, the temperature rise which was caused by the warm air striking the cold surface of the dew-point cup was reduced. Reference

to the detailed diagram of the dew-point cell in Fig. 4 shows how the size of the frost spot is kept at a minimum by bringing the small tip of the entering air-tube close to the cup.

Dew-point measurements were made on several dozen units, some of which were wetted deliberately, before it was decided to evaluate another method which will be discussed shortly. Actually, the dew-point results were encouraging as can be seen from the following two typical tests.

Test 1: The winding temperature was 228 F when the dew-point of the air from the shell was found to be 14 F. From the known weight of the cellulose insulation and the

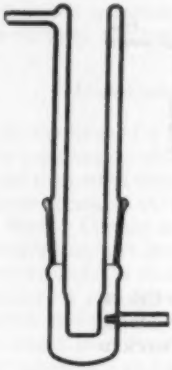

FIG. 4. . . .DEW-POINT CELL, FROST SPOT IS MINIMIZED BY BRINGING SMALL TIP OF ENTERING AIR TUBE CLOSE TO CUP

Piper curves a water content of 60 mg was calculated which may be compared with the later cold trap result of 55 mg.

Test 2: With a winding temperature of 223 F, a dew-point was measured at 21 F. The indicated water content was 90 mg as compared to the cold trap result of 55 mg.

The authors are skeptical about the close agreement between the dew-point and the cold trap results in Test 1. All of the water is not removed from a compressor unit during the 3-hr evacuation period of the cold trap test. Thus the result of this test should always be substantially lower than that of the preceding dew-point test on the same unit. A correction factor is needed on a short cold trap test whereas none is needed on the dew-point test which reports almost an equilibrium figure since but little water vapor is removed from the shell during the minute or so that is consumed by the test.

The major and perhaps only objection to the use of the dew-point method for estimating the moisture proved to be the time needed to get an accurate room temperature measurement of the resistance of the main windings. This measurement is essential for estimating the temperature of the windings, and thus of the insulation, at the time the air sample is removed for the dew-point check. It was found that an accurate cold resistance value could be measured only after a compressor had been stored in a constant temperature room for at least 12 hr.

This forced 12-hr delay in calculating the winding temperature (and this temperature is essential if the dew-point is to be used as a measure of moisture content) renders the dew-point method, as the authors have presently applied it, impractical as a production line quality control check. However, the authors have not given it up entirely and are planning to resume work with it at a later date. It is believed that if the compressor is placed in a small hot oven soon after it leaves the drying oven, the windings, shell and oven will shortly arrive at the same temperature and a dew-point test can be taken, and the results used without any need of resistance measurements of the windings. The details of such a modified test method, and the correlation between the dew-point tests and the moisture found by the old unmodified, reliable but slow cold trap test will require a substantial amount of experimental work.

USE OF CENTRIFUGE-COLD TRAP WITHOUT REHEATING

Only pessimistic conclusions can be drawn about the authors' efforts to omit the leak test and evacuate the compressors through a moisture detecting device as soon as possible after these had been removed from the dehydration oven. In these tests the loss of heat, with the resulting drop in temperature during the progress of the test, was slowed down by holding the compressor in the same foamed-resin box.

The results obtained with this simple modification of the original procedure were about as might be expected. The compressors with lower winding temperatures in general yielded less water in a 45-min cold trap test, that is seemed to be drier, than those which were evacuated when the windings were hotter. Later, however, these *colder*, and seemingly drier, units gave up more water to cold traps on reheating and testing than did the units which, when first tested, had windings which were closer to the correct test temperature of 300 F. Such results definitely showed that this form of the test was worthless for quality control.

EVACUATION OF THE UNCOOLED HOT COMPRESSOR

The next logical step was to place the hot compressor in a 300 F oven just as soon as it could be made ready after removal from the dehydration oven. This has been done and moisture test results have been obtained within one hour after compressors have been received. The water collected in a centrifuge-cold trap through which a compressor has been evacuated for 45 min is comparable with that collected after 3 hr by the old procedure. This conclusion is based on data obtained by continuing tests after the removal of the 45-min traps. These traps were removed without breaking the vacuum and evacuation of the compressors was continued through a second trap until the total evacuation time at 300 F equalled that called for in the original procedure. The water collected in a second trap averaged less than 10 percent of that found in a 45-min trap. Thus, the authors believe that this one-hour test method can be substituted directly for the older 3-hr method with only a slight change in the acceptance limits.

MOLECULAR SIEVE 4A AS A MOISTURE ESTIMATING REAGENT

Earlier the phrase *moisture detecting device* was used deliberately since many of the experiments, which were discussed in that section, were conducted with a commercial drier as the moisture collector rather than with a centrifuge-cold trap.

This drier was sized for about a 1-ton system and was charged with Molecular Sieve 4A as the desiccant. 4A is known to hold water much better under most conditions than several other desiccants which are used widely and it was hoped that an approximation to the water content of a compressor could be found by evacuating it through a freshly activated drier containing this material.

A number of tests were made with such a drier on an all glass system since a number of the variables could be controlled better in this than on a compressor.

For example, a freshly reactivated 4A drier was allowed to absorb a somewhat greater weight of water than is found in a compressor test. The weighed drier was then attached, by means of a short, heavy-walled piece of rubber tubing, to a glass system which had been previously filled with dry air. The glass system which had approximately the same volume as one of the compressors was then evacuated through the drier for 15 min after which the drier was removed and weighed. A weight loss of 1 mg was observed.

The drier was replaced on the still evacuated system and pumping was resumed through the drier for a total time of 2 hr. On reweighing an additional weight loss of 3 mg was noted.

This apparent error of 4 mg might be heartbreaking for a laboratory perfectionist but it is well within the tolerances of a test for dryness of compressors in a quality control laboratory. Moreover, while it is true that the observed apparent weight decreases may have been real, due to the pumping out of a trace of water or other gas from the desiccant, they could just as easily have been weighing errors. The drier was handled with hands, which had been washed but were ungloved. This means the moisture film on the surface of the copper was disturbed as was evidenced by the high loss of weight which was first observed when the drier was returned to the balance. Almost one-half hour may be needed at times before the weight becomes constant under such conditions of handling.

Other experiments have shown that, under the test conditions, vapors from the compressor lubricant are not retained by 4A and, carbon dioxide, which should interfere, is present in such small amounts that it can be ignored. Thus, the authors believe that either a cold trap or a drier which is charged with Molecular Sieve 4A can be used as the water collecting agent in the type of approximate quality control test which is under discussion.

COLD TRAP *vs* MOLECULAR SIEVE 4A

The centrifuge-cold trap avoids the use of a balance with the weighing delay which goes with it. However, the loss of moisture through improper handling has

TABLE 1. . . .WATER EVOLVED BY PAPER

TEMPERATURE	MILLIGRAMS OF WATER EVOLVED BY 75 GR OF PAPER	
	IN ONE YEAR	IN TWENTY YEARS
176	0.35	7
194	1.31	26
212	4.65	90
230	15.8	315
248	48.7	975
266	140.	2800

always been a problem. If dry ice is used as the coolant for the trap it may be present in inadequate amount or the trap may not be immersed to the proper level. Even if these details are properly looked after, entrained ice particles may be carried on through the trap by too high a mass flow of air during early stages of pumping. The weighing errors with the drier will lead to erratic results but the errors in the trap test will all be such as to indicate that the compressor under test is drier than it really is. This is definitely undesirable and thus the drier will probably prove to be the more useful tool in a factory.

SUMMARY

In this paper the need of a rapid and practical moisture procedure for checking the dryness of uncharged compressors has been stressed. A precise method, in which accuracy is achieved at the expense of speed or ease of control in the hands of laboratory technicians, is worthless for quality control purposes. A test which will measure the water content to the last milligram, even though the test is rapid, may not be as valuable as one which will give an approximate answer in less highly trained hands.

Moreover, why determine the presence of that last milligram of water when stalling of the motor on a low voltage line or blocking the access of cooling air to the condenser can cause the motor to overheat and lead to the generation of many times that weight of water by decomposition of some of the cellulose?

Lest the foregoing remarks be taken as expressing dissatisfaction with cellulose as a hermetic insulating material the authors disclaim any such opinion. They firmly believe that it is still the best and most practical sheet material which is presently available for use in this type of fractional horsepower refrigerating equipment. When an infusible substitute, which has the desirable physical and mechanical properties of cellulose is forthcoming, then, and only then, will this replace paper in this application.

It is true that overheating of cellulose will decompose it to form water as one of the products but the extent of this decomposition is negligible over a 10 or even a 20-year life when the motor is operated at anywhere near the design temperature. Many years ago, Dushman[3], using data reported from the Bell Laboratories, discussed the rates at which water is formed by heating paper to various temperatures. Table 1 has been modified by translating Dushman's figures into milligrams of water evolved per year by 75 g of paper. This is the weight of paper which might be used in a small hermetic motor which uses enamel on the wire and paper for other insulation. Centigrade temperatures have also been converted to Fahrenheit.

It can be observed from Table 1 that if a hermetic motor containing 75 g of cellulose insulation were designed to run continuously at 212 F, and did just that for 20 yr, 90 mg of water should be formed by cellulose decomposition during that period unless the thermal decomposition rate was modified by the presence of the oil or the refrigerant. (At 212 F, both the oil and Refrigerant 12 should be practically without action on cellulose.) Certainly, no manufacturer of a modern refrigerator or freezer would design such a device without adding a desiccant, or other substance, which would easily tie-up several times 90 mg of water.

A second unit might be designed never to run with a motor temperature below 230 F, and if this temperature were never exceeded, over 300 mg of water could be generated in 20 years without exceeding the capacity of a generously sized drier.

Many refrigerating compressor units are now being built with motors which are designed to operate at insulation temperatures of 240 F, or even higher, under pull-down conditions, or other heavy duty cycles which may temporarily exist under extreme weather conditions. However, the number of hours during which the temperature of the motor insulation is maintained at 240 F and above, will total only a small fraction of a year in a 20-year life. This assumes that the motor is not abused by being inadequately ventilated or otherwise improperly used by the owner. Hence, a good drier should always protect a modern refrigerator against damage by water, unless the insulation is abused to the point of destruction.

The use of a desiccant to protect against excessive amounts of water is stressed since domestic refrigerators and freezers now operate with such low temperatures at the point of expansion that moisture clogging can be a real problem in the absence of a good drier or the equivalent. The authors believe that the possibility of chemical damage by small amounts of water in these units has been greatly exaggerated particularly where Refrigerant 12 is the refrigerant and have seen little if any evidence which might cause them to fear small amounts of water in Refrigerant 22 systems.

The authors have dwelt at some length on this water problem since they disagree with the generally accepted opinion as to why the water content of a small, low temperature refrigerator must be controlled carefully. It is their opinion that mechanical clogging by ice is the real water problem in a clean and well-designed system. The weight of water must be definitely limited so that the capacity of the desiccant to hold this, and any additional water which may be generated through cellulose decomposition, will not be exceeded during a long, and frequently heavy-duty life. One way of avoiding careful quality control and the accompanying testing would be to use a drier with many times the capacity which might be needed under grossly abusive conditions. This is a needless expense and the added drier volume will upset the balance of the refrigeration cycle. Moreover, there is no drier made which will fit in the available space and yet have sufficient drying capacity to hold water which is dumped in by the cupful.

There is no substitute for a good system of checks and tests for the control of quality. The authors hope that this discussion of practical tests for estimating the dryness of uncharged compressors on a production line will prove useful in maintaining the quality and good reputation of this type of equipment.

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[2] J. D. Piper: Moisture equilibrium between gas space and fibrous materials in enclosed electric equipment (*American Institute of Electrical Engineers Transactions*, Vol. 65, 1946, p. 791).

[3] S. Dushman: *Scientific Foundations of Vacuum Technique* (John Wiley & Sons, Inc., 1949, p. 545).

DISCUSSION

R. T. DIVERS, Syracuse, N. Y.: When the fact was mentioned that the water would reside in the drier, was it not the real meaning that there would be a state of dynamic equilibrium present where there would be some water in the discharge gas from the drier?

Is the reference to a molecular sieve drier?

F. L. TARLETON, Chicago, Ill.: Reference was made to Refrigerant 12. Do the authors have any experience with a molecular sieve with Refrigerant 22?

J. P. HURTGEN*, Holland, Mich.: How does one go about setting the limits of estimated moisture content? In other words, to what is the estimate of the water content related and how does one determine whether it is good or bad? The question is asked to learn what criteria were used, not to learn the factors themselves.

Mention was made of plotting the moisture content. Was this the circulating moisture?

AUTHORS' CLOSURE (Mr. Elsey): Mr. Divers is correct in his comment. In a Refrigerant 12 system the desiccant in question holds water so tightly that most of the water in the system is in the molecular sieve drier used.

With reference to the question asked by Mr. Tarleton, the authors have no experience with a molecular sieve with Refrigerant 22. The authors have urged extensive field testing of refrigerators which are equipped with molecular sieve driers and charged with Refrigerant 22. However Refrigerant 12 has met the needs of the engineers in systems which need a permanent drier and they have hesitated to experiment on a large scale with Refrigerant 22.

Some of the large air-conditioning systems which use Refrigerant 22 are equipped with driers in which molecular sieve is incorporated in the desiccating block until the systems are thoroughly dried but in general the driers are removed later. Where the molecular sieve has been used on such systems no detrimental effects have been observed through reaction with the refrigerant.

There is some tendency to refuse to use a combination of Refrigerant 22 and molecular sieve on the basis of laboratory tests. It seems to the author that no material which offers improved performance should be rejected because of poor behavior in an accelerated tube test. Frequently the acceptance of excellent materials has been delayed for years because a sample looks bad in a tube under conditions which bear no real relation to the conditions in an actual refrigerating system.

After a bad result in a test tube one should figure out a test in an actual practical refrigerating system and determine the rate at which the material under test will yield harmful substances in sufficient quantity to interfere with the performance of the machine. If the new substance will materially reduce the useful life of a machine then it is bad. If it does not, then it is worth using. One of the disadvantages of a laboratory is the pessimism about using new materials which comes out of most reports. Some times a laboratory helps but very frequently it is a definite hindrance.

Mr. Hurtgen's first point has to do with highly secret information which the engineers withhold even from their own quality control department and it can hardly be disclosed here.

Actually hot compressor units have been evacuated for approaching 100 hours and the rate at which the moisture is pulled out during this evacuation has been plotted. After this was done on a number of units it was possible to correlate the weight of water, which was pulled out in a definite time during the first few hours of the evacuation, with the total water which was pulled out in the long time evacuation. Instead of explaining the details to the shop the practice is to merely specify the procedure to be used during a definite test period and to set a moisture limit for this test. It is known that there is more water which could be pulled out but the tests with units establish the fact that the limits chosen are conservative for the units in question. It must be remembered that the geometry of the system, and the nature and quantity of the insulation as well as of the refrigerant, and the service conditions really set the moisture limit which can be tolerated.

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* General Electric Company.

There are certain types of paper and other forms of cellulose which have been used for years. Experience establishes how rapidly these materials will give up water and regain it. The prior heat treatment of cellulose will alter the amount of water it will hold. Thus the choice of the insulation, the nature of its processing and every feature of the unit design enter into the setting of the tolerable water limit.

With regard to Mr. Hurtgen's second question the tests in question were on uncharged compressor units. These were conditioned in air of known relative humidity and were then dried under definite conditions. The water which was removed by evacuation was collected in a series of weighed traps and the weight of water removed was plotted against the time of evacuation.

In the tests which were discussed in this paper the indicated procedure was followed in each case following which the unit was reheated and evacuated through cold traps and the water collected and estimated.

1693

No. 1693

EFFECT OF PRESSURE AND TEMPERATURE ON THE VISCOSITY OF REFRIGERANTS IN THE VAPOR PHASE

By C. Z. KAMIEN* AND O. W. WITZELL**, WEST LAFAYETTE, IND.

MEASUREMENTS made of the viscosity of nine fluorinated hydrocarbon vapors, Refrigerant 12 (CCl_2F_2), Refrigerant 13 ($CClF_3$), Refrigerant 14 (CF_4), Refrigerant 21 ($CHCl_2F$), Refrigerant 22 ($CHClF_2$), Refrigerant 23 (CHF_3), Refrigerant 114 ($CClF_2$-$CClF_2$), Refrigerant 115 ($CClF_2$-CF_3), and Refrigerant C318 (Cyclic C_4F_8) were carried out in a Hoeppler type viscometer with a pyrex tube and ball. These covered a temperature range of about 30 to 90 C and pressures up to 20 atmospheres. Four isotherms were used in the measurement of the effect of pressure on the viscosity of refrigerant vapors.

The viscosities of some fluorinated hydrocarbon vapors were measured using a rolling-ball viscometer. The latter consists of a ball rolling down an inclined tube filled with the fluid under investigation. An examination of the variables affecting the ball velocity in such a tube indicates that

$$\mu = f(\rho_f, \rho_B, D_t, D_B V, \alpha) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

μ = dynamic viscosity.
ρ_f = density of the fluid.
ρ_B = density of the ball.
D_t = internal diameter of the roll tube.
D_B = diameter of the ball.
V = ball velocity.
α = angle of inclination of the tube.

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A dimensional consideration of the variables involved shows that Equation 1 can be written as

$$\phi\left[\frac{D_B^3\rho_f \sin\alpha}{V\mu}, \frac{\rho_B}{\rho_f}, \frac{D_B}{D_t}\right] = 0 \quad \ldots\ldots\ldots (2)$$

Hubbard and Brown[1] have determined the form of the function ϕ for a system consisting of a closed straight circular tube with a closely fitting ball. The fluid is transpired through the crescent-shaped space between the ball and the tube walls when the ball rolls down the tube. Their analysis showed that if laminar flow is maintained the following equation is applicable to the rolling ball viscometer

$$\mu = K\theta(\rho_B - \rho_f)\sin\alpha \quad \ldots\ldots\ldots (3)$$

In this relation K is the calibration constant for the viscometer and includes the ball and tube diameters, and the length of the tube, and θ represents the time for the sphere to roll from upper to lower gage mark. The calibration constant is determined by comparison with a fluid whose viscosity is known.

Equation 3 was employed in calculating the viscosities of the fluorinated hydrocarbons reported herein.

The effect of temperature changes on the calibration constant is appreciable when high temperatures are encountered or when the materials used have high coefficients of expansion. For a given temperature the effect is greater if the ball and tube are of different materials. An increase in temperature increases the diameter of the ball and the tube. If the ball and tube are made of the same material, which is the case in these measurements, there will be no change in the diameter ratio when the temperature is increased. Since the calibration constant involves not only the diameter ratio but also depends on the actual diameters, a change in temperature will affect the calibration coefficient. Therefore the values of the coefficient had to be determined whenever temperature changes, physical changes of apparatus, or reassembling occurred.

In 1933, Hoeppler[2] reported the results of his experimental work on the viscosity of air, carbon dioxide, and oxygen. He used a glass tube at an angle of 80 deg with the horizontal, a ball of about ¾-in. diam., and calibrated the viscometer with water.

The first reported measurements of the viscosity of fluorinated hydrocarbon vapors were made in 1939 by Benning and Markwood[3]. They used a rolling ball viscometer with a tube at 80 deg angle of inclination. Measurements were made at pressures from 1 to 9 atmospheres. Measurements of viscosity were obtained for the vapor phase of Refrigerant 11, Refrigerant 12, Refrigerant 21, and Refrigerant 22. Measurements for Refrigerant 113 were made only at 0.1 and 0.3 atmospheres. The temperature range of these measurements was 0 to 80 C.

In 1945, Graham[4] used the Benning and Markwood equipment to measure the viscosity of Refrigerant C318 in the liquid and vapor phases over the temperature range of 0 to 70 C and 9 to 100 C respectively. A maximum pressure of 10 atmospheres was obtained in the measurements of vapor viscosities. In further work, Graham[5] determined theoretically the vapor viscosity of Refrigerant 12 at high pressures and temperatures. He extrapolated the values obtained experimentally by Benning and Markwood at low temperature to obtain initial values

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[1] Exponent numerals refer to References.

for the use of the correlations with reduced coordinates suggested by Uyehara and Watson[6].

Measurements of Refrigerant 22, Refrigerant 23, and Refrigerant 115 vapor viscosities for a temperature range of −36 to +60 C were obtained by Coughlin[7] in 1953. These measurements were made at atmospheric pressure with a modified Hoeppler type unit with angle of inclination of about 15 deg, to reduce the velocity of the rolling ball and consequently the Reynold's number.

The effect of pressure upon the viscosity of 3 refrigerant compounds was reported by Makita[8] in 1954. The viscosity was measured at the following conditions: Refrigerant 12, 25 to 200 C and 1 to 16 atm., Refrigerant 21, 25 to 200 C and 1 to 19 atm. The apparatus used in this investigation was a rolling-ball viscometer consisting of a soda glass tube and a steel ball. The inner diameter of the tube was 5.5-6.0 mm, and the tube could withstand 240 atm. inner pressure. The ball was returned to its initial position by means of a magnet. The viscometer tube was submerged in an electrically heated thermostat regulated to 0.1 C. The viscometer was calibrated using carbon dioxide. Wellman[9] reported measurements in 1955 made with the same viscometer used in the present investigation. Measurements were made of the viscosities of 4 vapors, namely, Refrigerant 12, Refrigerant 13, Refrigerant 22, and Refrigerant 114. The measurements covered a temperature range of 3 to 90 C. For two of the vapors, viscosity measurements were made at pressures up to 20 atm. Wellman's results were in agreement with the kinetic theory relation for dilute gases in which the viscosity is a linear function of the square root of the absolute temperature, or

$$\mu = A\sqrt{T} - B \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where A and B are constants and T is the absolute temperature.

APPARATUS

Apparatus used in this investigation is described in detail by Wellman and is a modification of the Hoeppler-type unit and of that employed in earlier work on the viscosity of refrigerant gases. Details of the viscometer body assembly can be found in the work of Wellman.

The ball and tube were made of pyrex glass and the dimensions were as follows:

Ball diameter	1.59055 cm.
Tube diameter	
at upper gage mark	1.59360 cm.
at lower gage mark	1.59350 cm.
Distance between gage marks	10.0 cm.
Over-all length of tube (approx.)	20.7 cm.
Density of ball at 20 C	2.2291 g/cm³

An additional gage mark on the tube was located midway between the upper and lower gage marks.

The viscometer was completely enclosed in a heating jacket to maintain it at the desired temperature. Glass windows were installed in rectangular slots in the heating jacket in the front and back panels to permit observation of the tube and ball. The temperature of the viscometer was maintained constant by circulating water from a 10-gal reservoir through copper tubes surrounding the viscometer. The reservoir was kept at constant temperature by electrical heaters

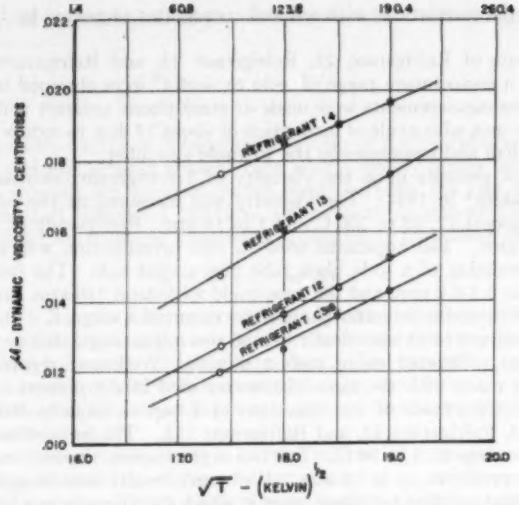

FIG. 1. . . .VAPOR VISCOSITIES OF REFRIGERANTS 12, 13, AND 14, AND C318 AT ONE ATMOSPHERE

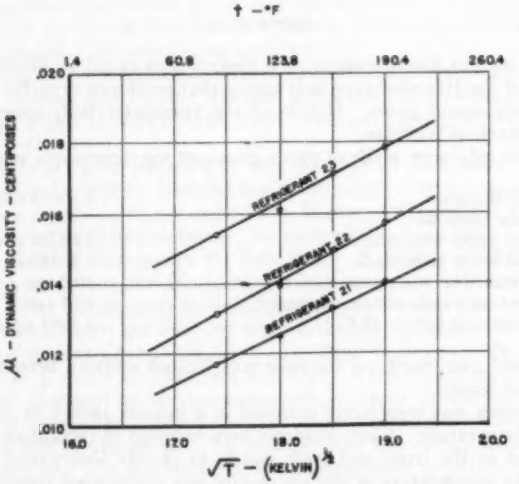

FIG. 2. . . .VAPOR VISCOSITIES OF REFRIGERANTS 21, 22 AND 23 AT ONE ATMOSPHERE

and a water cooling coil. It was possible to hold the temperature in the viscometer within ±0.05 C during any run.

The duration of roll was obtained by means of an electric timer.

CALIBRATION

Since the rolling-ball viscometer is a secondary viscometer, it had to be calibrated with a fluid of known viscosity. Air was used as the calibrating fluid for atmospheric pressures, as it was readily available and the viscosity was known. For pressures higher than atmospheric, nigrogen was used as the calibrating fluid

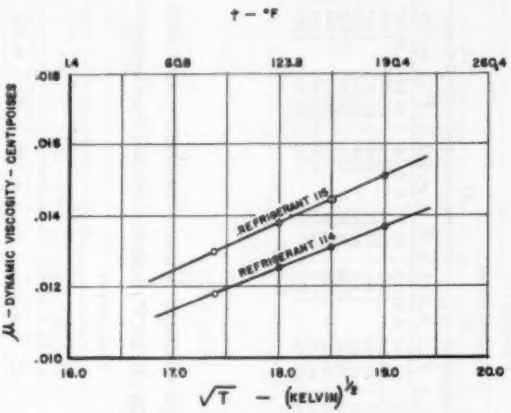

FIG. 3. . . .VAPOR VISCOSITIES OF REFRIGERANTS 114 AND 115 AT ONE ATMOSPHERE

because it was easily obtainable in a highly purified state and the viscosity was also known. The viscosities of air and nitrogen used in the calibrations were obtained from the NBS tables[10, 11].

After the viscometer was brought to the desired temperature, a vacuum pump evacuated the system, reducing the pressure to about 1 mm of mercury absolute pressure. After evacuation of the system, air was introduced from the atmosphere. The air was dried and purified by passing it through a calcium chloride drying tube, a sintered glass filter, a drying tube charged with P_2O_5, a second glass filter, and an oil trap.

The nitrogen used for calibration at pressures higher than atmospheric was introduced into the viscometer after evacuating the system and flushing it through with nitrogen before the experimental sample was admitted. The sample was put into the viscometer at the highest pressure desired (20 atm.) and then reduced to the lowest pressure desired (3 atm.), with readings being taken at various pressures between these extremes.

The refrigerants used in this work were high-purity samples of the respective refrigerants. All were specially dried and deaerated and had a purity of 99.9+

TABLE 1. . . .EXPERIMENTAL DATA OF VAPOR VISCOSITIES AT 1 ATM.

(tC = temperature, Centigrade; μ(CP) indicates viscosity, centipoises)

REFRIGERANT																	
12		13		14		21		22		23		114		115		C318	
tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)
30.19	0.01274	30.19	0.01488	30.00	0.01762	29.98	0.01170	29.84	0.01316	29.77	0.01545	29.99	0.01162	30.19	0.01291	30.19	0.01202
30.23	1274	29.89	1480	29.93	1762	30.00	1172	30.13	1317	29.95	1545	29.88	1160	29.98	1291	30.13	1202
49.48	1358	49.50	1605	49.76	1868	49.61	1251	49.60	1391	49.75	1609	49.81	1249	49.61	1376	49.58	1266
49.56	1362	49.50	1601	49.74	1868	49.60	1249	49.66	1391	49.76	1609	49.81	1250	49.59	1376	49.66	1266
70.14	1436	69.93	1638	69.95	1903	70.02	1328	70.08	1500	70.14	1719	70.10	1300	70.23	1442	70.02	1364
70.00	1438	69.93	1635	70.01	1903	70.00	1331	70.03	1500	70.07	1719	70.22	1304	70.15	1442	70.10	1364
89.95	1525	89.99	1769	89.98	1971	89.98	1399	89.97	1572	89.98	1787	89.98	1363	90.00	1505	89.99	1434
89.96	1523	89.99	1769	89.97	1971	89.98	1399	89.97	1572	89.97	1787	89.99	1362	90.00	1505	89.98	1434

TABLE 2. . . .EXPERIMENTAL DATA OF VAPOR VISCOSITIES AT 3 ATM.

(tC = temperature, Centigrade; μ(CP) indicates viscosity, centipoises)

REFRIGERANT																	
12		13		14		21		22		23		114		115		C318	
tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)
30.00	0.01724	30.00	0.01500	29.99	0.01765			30.00	0.01323	30.01	0.01539						
49.99	1378	50.00	1615	49.98	1842			50.00	1416	50.00	1625			50.01	0.01387		
70.00	1453			70.00	1914	70.00	0.01340	69.99	1516	70.03	1733	69.99	0.01305	70.00	1455	69.99	0.01377
89.99	1542			89.98	1976	89.96	1410			90.01	1802	89.97	1371			90.00	0.01449

TABLE 3. . . .EXPERIMENTAL DATA OF VAPOR VISCOSITIES AT 6 ATM.

(tC = temperature, Centigrade; μ(CP) indicates viscosity, centipoises)

REFRIGERANT																	
12		13		14		21		22		23		114		115		C318	
tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)
29.98	0.01283	30.01	0.01511	29.99	0.01770			29.98	0.01331	30.00	0.01543						
49.98	1414	49.99	1629	50.00	1852			50.01	1434	50.00	1638			50.01	0.01395		
70.00	1478			70.00	1926	70.00	0.01363	70.01	1541	70.00	1745	70.00	0.01317	70.01	1468	70.00	0.01404
90.00	1572			89.98	2015	89.94	0.01424			89.99	1821	89.97	1382			89.99	1487

TABLE 4. . . .EXPERIMENTAL DATA OF VAPOR VISCOSITIES AT 12 ATM.

(tC = temperature, Centigrade; μ(CP) indicates viscosity, centipoises)

REFRIGERANT													
12		13		14		22		23		115		C318	
tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)
		30.00	0.01551	29.98	0.01807			30.00	0.01565				
70.01	0.01554	50.01	1680	49.98	1872	50.00	0.01465	50.00	1681	50.00	0.01425		
90.00	1637			70.00	1962	70.00	1592	70.01	1768	70.00	1508		
				89.98	2061			89.98	1894			89.99	0.01585

TABLE 5. . . .EXPERIMENTAL DATA OF VAPOR VISCOSITIES AT 20 ATM.

(tC = temperature, Centigrade; μ(CP) indicates viscosity, centipoises)

REFRIGERANT											
12		13		14		22		23		115	
tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)	tC	μ (CP)
		30.00	0.01618	29.96	0.01860			29.99	0.01623		
		50.00	1759	50.00	1951			50.00	1773		
				70.00	2049	70.00	0.01684	69.99	1853	70.01	0.01609
90.00	0.01745			89.97	2147			89.98	1966		

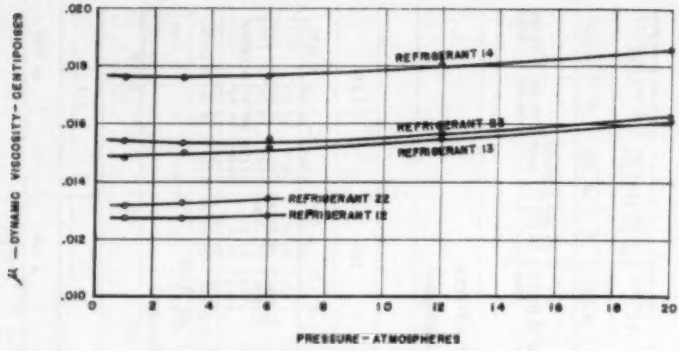

FIG. 4. . . .PRESSURE EFFECT ON REFRIGERANT VAPOR VISCOSITIES AT 30 C (86 F)

percent. The Refrigerant 114 sample contained approximately 8 percent of the isomer, Refrigerant 114a (CCl_2F Cl_3).

DISCUSSION OF RESULTS

The viscosities at atmospheric pressure are shown graphically in Fig. 1 (Refrigerants 12, 13, 14, and C318), Fig. 2 (Refrigerants 21, 22, and 23), and Fig. 3 (Refrigerants 114 and 115).

Benning and Markwood showed that a satisfactory correlation can be achieved for the viscosity temperature relation by considering the viscosity as a linear function of $\sqrt{T}$. The viscosities are therefore plotted versus $\sqrt{T}$ where T is

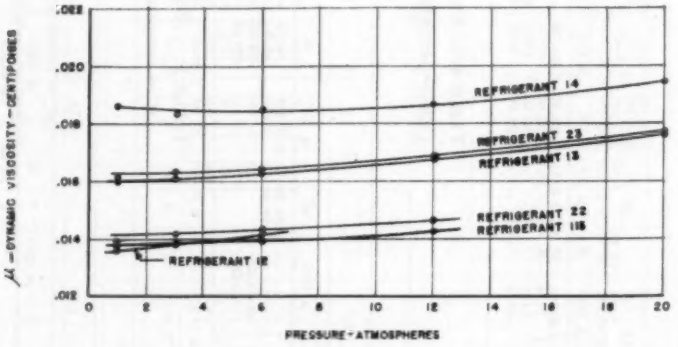

FIG. 5. . . .PRESSURE EFFECT ON REFRIGERANT VAPOR VISCOSITIES AT 50 C (122 F)

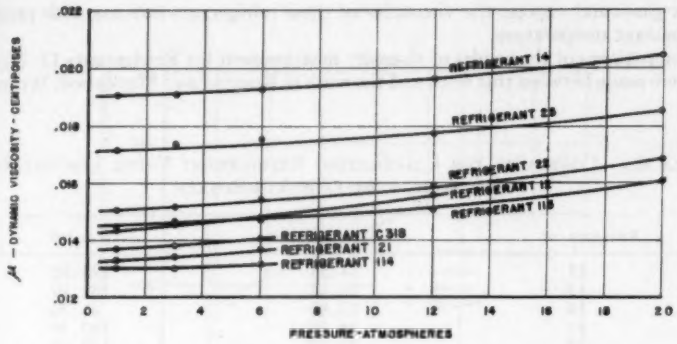

FIG. 6. . . .PRESSURE EFFECT ON REFRIGERANT VAPOR VISCOSITIES AT 70 C (158 F)

the absolute temperature in deg K and μ is the dynamic viscosity in centipoises. The experimental results can be represented by a straight line as shown by Equation 4.

The experimental data obtained during this investigation are listed in Tables 1 to 5.

Table 6 lists the constants required to calculate the viscosities of the compounds studied in the vapor phase; and Table 7 lists the viscosities in centipoises of the refrigerants in the vapor phase at 20 F intervals and at atmospheric pressure as calculated from the equation and the appropriate constant.

Figs. 4, 5, 6 and 7 show the effect of pressure on viscosity at 30, 50, 70 and 90 C respectively. In accordance with data on pressure dependence of viscosity for

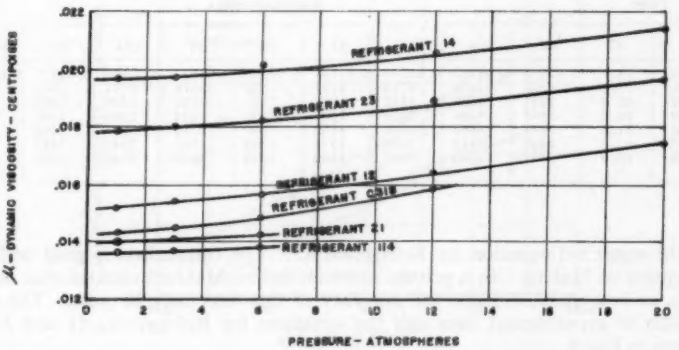

FIG. 7. . . .PRESSURE EFFECT ON REFRIGERANT VAPOR VISCOSITIES AT 90 C (194 F)

other gases and vapors, the viscosities of these refrigerants increase with pressure at constant temperature.

Comparisons of the results of viscosity measurement for Refrigerants 12, 21, and 22 were made between this work and the work of Benning and Markwood, Wellman,

TABLE 6. . . .CONSTANTS FOR CALCULATING REFRIGERANT VAPOR VISCOSITIES BY EQUATION 4 AT ONE ATMOSPHERE

REFRIGERANT	$A \times 10^4$	$B \times 10^4$
12	14.55	125.80
13	16.15	131.10
14	12.80	48.30
21	14.75	140.30
22	16.20	150.60
23	15.30	111.80
114	12.50	101.30
115	13.40	104.00
C318	14.80	137.70

Coughlin and Makita. In determining the calibration constant, Benning and Markwood did not consider the variation with temperature of the constant due to changes in the annular area between the tube and the ball. The viscosities reported by Benning and Markwood were recalculated using the same air viscosities as in this work. Fig. 8 shows a comparison of the data of the various investigators

TABLE 7. . . .REFRIGERANT VAPOR VISCOSITIES AT ONE ATMOSPHERE

(*Viscosity in Centipoises*)

TEMP		REFRIGERANTS								
F	C	12	13	14	21	22	23	114	115	C318
80	26.7	0.01258	0.01481	0.01730	0.01147	0.01295	0.01527	0.01148	0.01277	0.01182
100	37.8	1307	1536	1774	1197	1350	1579	1191	1322	1232
120	48.9	1351	1585	1812	1242	1401	1625	1220	1363	1277
140	60.0	1397	1636	1853	1289	1451	1674	1268	1406	1324
160	71.1	1441	1685	1891	1333	1499	1720	1306	1446	1368
180	82.2	1483	1732	1929	1376	1546	1765	1342	1485	1411
200	93.3	1527	1780	1967	1420	1595	1810	1380	1525	1456

to the suggested equation for Refrigerant 12. The comparison is good with the exception of Makita. In a private communication, Makita indicated that on the basis of new measurements, the accuracy of this data may be poor. The comparison of experimental data and the equations for Refrigerants 21 and 22 are shown in Fig. 9.

Further work is being carried out to determine the viscosities of refrigerants beyond the temperatures considered in this investigation. The new values of

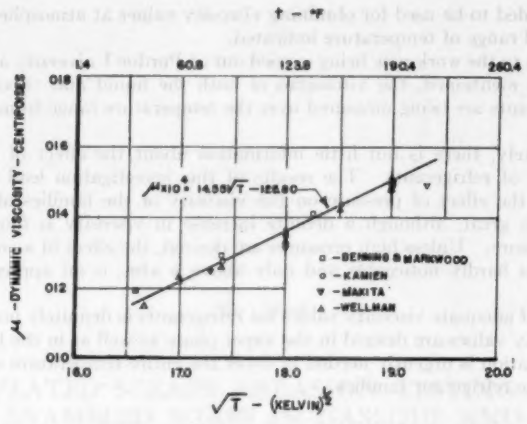

FIG. 8. . . .COMPARISON OF RESULTS FOR REFRIGERANT 12 VAPOR VISCOSITY MEASUREMENTS AT ONE ATMOSPHERE

viscosity obtained tend to confirm Equation 4 and the constants cited in Table 6. These new values cover the range of temperatures from 90 to 200 C and show little variation from the equation developed in this work. Equation 4, therefore, can

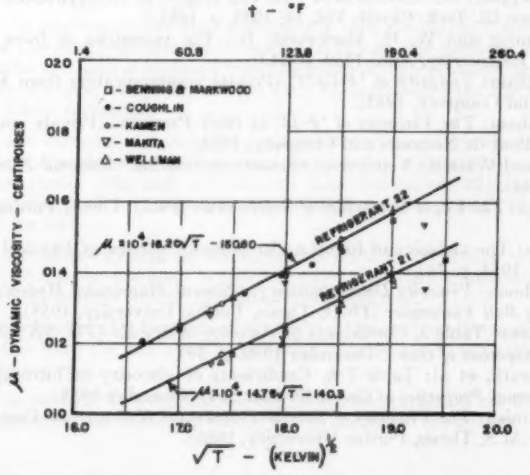

FIG. 9. . . .COMPARISON OF VAPOR VISCOSITIES FOR REFRIGERANTS 21 AND 22 AT ONE ATMOSPHERE

be recommended to be used for obtaining viscosity values at atmospheric pressure over the total range of temperature indicated.

In addition to the work now being carried out at Purdue University at the higher temperatures mentioned, the viscosities of both the liquid and vapor phases of these refrigerants are being measured over the temperature range from −100 F to 300 or 400 F.

Unfortunately, there is but little information about the effect of pressure on the viscosity of refrigerants. The results of this investigation lead to the conclusion that the effect of pressure on the viscosity of the families of refrigerant vapors is not great, although a definite increase in viscosity is found with increasing pressure. Unless high pressures are desired, the effect of a small increase in pressure is hardly noticeable and only above 6 atm. is an appreciable effect noticed.

The lack of adequate viscosity values for refrigerants is definitely noticeable and more viscosity values are desired in the vapor phase as well as in the liquid phase. More information is urgently needed to cover the entire temperature and pressure ranges for the refrigerant families.

ACKNOWLEDGMENT

The authors acknowledge the help of the Freon Products Division, E. I. duPont de Nemours and Co., Inc., in both academic and financial matters to make this investigation possible.

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1694

No. 1694

REPEATED SCRAPE ABRASION TESTING OF ENAMELED WIRES IN GASEOUS AND LIQUID REFRIGERANTS

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DEVELOPMENT of new and better wire insulation for hermetic applications requires quick test methods which approximate or simulate equipment conditions in actual operation. Numerous test methods have been proposed and are used for evaluating magnet wire insulations for hermetic motors. Some of these are:

Blister test: In one test of this type a sealed hot tube containing the enameled wire specimen and the liquid refrigerant is broken to release the pressure suddenly. In another, the wire specimen is removed from the liquid refrigerant and placed quickly in an oven at an elevated temperature (250 F). In both tests, the enameled wire is examined for blister formation.

Sealed tube test: Stressed and unstressed enameled wire specimens along with system components, including the liquid refrigerant, are sealed in a glass tube and aged at an elevated temperature. The tube contents are inspected periodically for components degradation as evidenced by enamel crazing, oil darkening, or deposition of extracted solids on the tube walls.

Thumbnail test: After exposing the wire specimen for a period of time to the liquid at a given temperature, the specimen is removed from the sealed container and scraped with the thumbnail to determine if the insulation has softened.

Alcohol-toluol extractables: Low-molecular-weight, unreacted, or decomposition materials are removed from the insulation by refluxing and the resulting weight loss of the insulation is determined.

Refrigerant extractables: This test method determines the amount of the wire insulation that goes into solution due to the solvent action of the liquid refrigerant.

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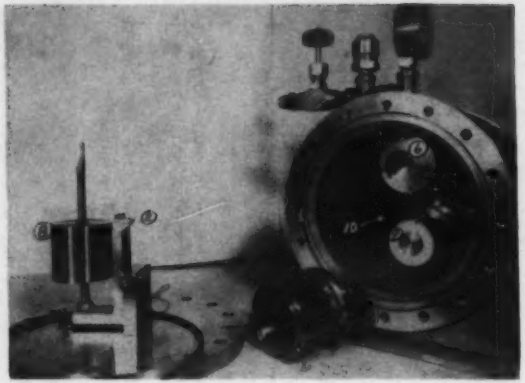
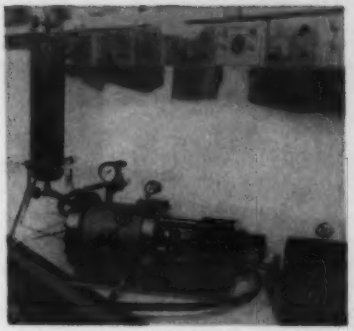
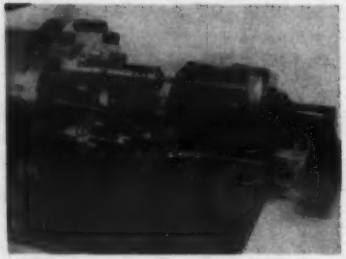


(1) multiple wire specimen holder; (2) weighted scrape head; (3) lower shaft; (4) angular and axial positions of lower shaft; (5) upper shaft; (6) raise-lower cam; (7) pair of pins; (8) weight cylinder; (9) teflon cylinders; (10) cantilever spring; (11) drive motor and stroke counter; (12) high-pressure pyrex glass viewing ports





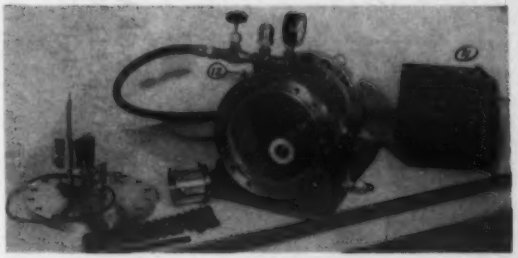


FIG. 1. . . .REPEATED SCRAPE ABRASION TESTER

Hot-cold surface deposit test: This test is based on the amount of material extracted by an oil-refrigerant mixture from an enameled wire, which subsequently varnishes-out on a hot surface or precipitates on a cold surface.

Of these, only the thumbnail test is intended to measure the softening of the enamel by the refrigerant. The ability of the enamel to withstand mechanical abrasion is important in some applications. Experience has shown that failure rates are higher for certain integral horsepower units than for fractional horsepower units. This may be due in part to the adverse mechanical conditions which exist in an application of this type. Adverse conditions include high stresses in the windings, motion of the wires and temperature effects. The end windings of the larger motors extend outward from the iron core a greater distance than those of small motors, thereby subjecting them to higher mechanical stresses.

Because a softened insulation is more susceptible to mechanical damage under adverse conditions, it seemed desirable to design a repeated scrape abrasion tester which would permit testing enameled wire specimens under load while immersed in a refrigerant gas or liquid. This would simulate actual conditions of operation in hermetic systems. It is important that the softening effect be measured in the presence of the refrigerant since the rate of recovery of most softened materials is rapid after removal from the refrigerant atmosphere. A number of developmental wire enamels have been tested in a tester of this general type. The principal advantages of the authors' own newly designed tester are that 48 tests can be made on 16-wire specimens in a single charging of the tester, and that the load can be changed from outside the tank under full refrigerant pressure.

DESCRIPTION OF NEW TESTER

This apparatus was designed for testing the repeated scrape abrasion resistance of enameled magnet wires in the presence of refrigerant liquids and gases at various temperatures and pressures.

The unit is shown in accompanying illustrations (Fig. 1). A multiple wire specimen holder (1) and a weighted scrape head (2) with a transverse 28-mil steel needle for abrading the wire insulation are mounted suitably in a hermetically-sealed stainless steel chamber. Adjustments of specimen position, scrape head position and scrape head load are made from outside the chamber under full gas pressure by means of two shafts sealed by neoprene O-rings (a teflon O-ring is now being considered since the neoprene O-ring on the lower shaft wears rather rapidly during tests in liquid refrigerants). The lower shaft (3) also serves to drive the specimen holder in a reciprocating motion, ⅜-in. stroke, under the scrape head.

The specimen holder is a drum on which 16 wires, approximately 3 in. long, can be mounted. Each wire can be tested at two different positions along its length. Thus, it is possible to obtain 32 test values on any specific wire type, two values on each of 16 different types of wires, or any combination within these two extremes, with a single loading and charging of the tester. The tester is now being modified to provide three instead of two test positions on each of the 16 wire specimens. In addition to increasing the number of test areas for a single charging of the tester, this will enable testing the same wire specimens under three conditions, such as in air, refrigerant gas and refrigerant liquid.

Any one of the test areas can be positioned under the scrape head by adjustment of the angular and axial positions (4) of the lower shaft, which carries the specimen holder. The scrape head is raised during positioning of the specimen holder, then

lowered gently to rest on the specimen to be tested next. This is done by rotating the upper shaft (5) which carries the raise-lower cam (6). Also, the load on the scrape head can be adjusted over a wide range from outside the chamber. This is done by pushing the upper shaft into the chamber far enough to allow the pair of

TABLE 1—REPEATED SCRAPE ABRASION RESISTANCE OF VARIOUS TYPES OF WIRE ENAMELS IN AIR AND REFRIGERANT 22 GAS AND LIQUID

(Room Temperature—740 Gram Load—0.028 in. Diam Needle)

ENAMEL	TYPE	WIRE SIZE AWG	BUILD, MILS	AGING TIME, HR	REFRIGERANT PRESSURE, PSI	[a] REPEATED SCRAPE ABRASION		
						IN AIR CYCLES	IN R22 GAS CYCLES	IN R22 LIQUID CYCLES
formvar	phenolic modified polyvinylacetal	18	3.0	68	125	37	4	—
		17	4.0–4.1	22	125	165	7[b]	—
		17	3.0	22	120	246	—	2
A	polyacrylonitrile	17	2.7–2.8	68	125	102	14	—
		17	2.7–2.8	24	135	126	—	4
B	epoxy modified polyester	17	2.6–2.7	20	125	38	43	—
C	medium functionality terephthalate polyester	18	2.4–2.5	20	125	52	26	—
			2.4–2.5	20	135	75	—	5
D	medium functionality terephthalate polyester	17	3.2	68	125	74	37	—
		17	3.2	20	125	117	40	—
		17	3.2	20	135	146	—	7
E	polyamide	17	2.3–2.5	24	125	41	177	—
		17	2.3–2.5	20	125	50	—	275
F	amine catalyzed epoxy	17	3.3	68	125	28	32	—
		17	2.8–3.0	22	125	32	33	—
		17	2.8–3.0	22	125	25	—	6
		16	5.0	22	135	611	—	163
G	formvar modified epoxy	17	3.0–3.1	22	121	20	10	—
		17	3.0–3.1	22	120	31	—	4
H	polyesteramide	17	3.4	68	125	22	39	—
		17	2.5–2.6	22	120	35	—	3
I	polyurethane modified polyesteramide	17	2.9–3.0	24	135	136	—	310

[a] Average of three values.
[b] Average of nine tests.

pins (7) near the outer edge of the cam disk to engage the corresponding holes in the face of the load weight cylinder and then rotating the shaft to turn the weight cylinder (8) on its adjusting screw thread. One complete revolution of the knob results in a 20-gram change in the application of the scrape head load.

The specimen holder is mounted on a teflon cylinder (9) in order to insulate it from the chamber and scrape head, but it is in sliding contact with a cantilever spring (10) mounted from a through-bushing on one end plate. The specimen wires are supported in grooves, passing through holes in end rings on the drum, and are clamped tightly in one end ring by set screws. These set screws puncture the wire enamel and connect the copper wire electrically to the specimen holder.

The failure point is indicated electrically when electrical contact is established between the scrape head needle and the copper wire through the abraded insulation, automatically stopping the drive motor and stroke counter (11).

There are 2 high-pressure pyrex glass viewing ports (12) in the chamber wall, through which the operation can be observed.

Not shown are a dial thermometer and a thermocouple mounted in the end plate of the tester and extended to the center of the unit near the test samples. The unit can be heated by means of a heating tape, wound around the outside of the cylinder, and the temperature controlled by a temperature controller within ±1 C.

TABLE 2—REPEATED SCRAPE ABRASION RESISTANCE OF VARIOUS ENAMELS IN AIR AND REFRIGERANT 12 GAS

(Room Temperature—740 Gram Load—0.028 in. Diam Needle)

ENAMEL	TYPE	WIRE SIZE AWG	BUILD, MILS	AGING TIME, HR	REFRIGERANT PRESSURE, PSI	*REPEATED SCRAPE ABRASION		
						IN AIR CYCLES	IN R12 GAS CYCLES	IN R12 LIQUID CYCLES
formvar	phenolic modified polyvinylacetal	17	2.3–2.5	24	80	75	42	—
B	epoxy modified polyester	17	2.6–2.7	20	80	36	89	—
C	medium functionality terephthalate polyester	18	2.4–2.5	23	80	57	148	—
D	German medium functionality terephthalate polyester	17	3.2	22	80	75	191	—
E	polyamide	17	2.3–2.5	22	80	50	143	—
H	polyesteramide	17	2.3–2.4	20	75	13	35	—
I	polyurethane modified polyesteramide	17	2.9–3.0	24	80	51	107	—

* Average of three values.

Although the tester shows effects of refrigerants on enameled wires, there are rather wide variations in results of tests made under the same conditions. Similar variations have been experienced in the use of the standard repeated scrape abrasion tester. A higher degree of consistency and reproducibility is desired. It appears that at times some enamel adheres to the scrape needle and tends to increase the scrape values. This difficulty has been eliminated in part by stroking the needle over a fine brush or wooden insert mounted in one of the wire specimen positions on the drum before testing each specimen.

TEST RESULTS

Tables 1 and 2 give average repeated scrape abrasion values for various chemically different wire enamels as tested in air and Refrigerants 12 and 22 (gases and liquids). Tables 3 and 4 show results of tests using smaller loads and needles of reduced diameters.

TABLE 3. . . .REPEATED SCRAPE ABRASION RESISTANCE OF VARIOUS TYPES OF WIRE ENAMELS IN AIR, REFRIGERANT 22 GAS AND LIQUID

(Room Temperature—450 Gram Load—0.016 in. Diam. Needle—Average of 4 Values in Cycles)

ENAMEL	BUILD (MILS)	AIR	AFTER 22 HR IN R22 GAS	RECOVERY AFTER 22 HR IN AIR	AIR	AFTER 22 HR IN R22 LIQUID	RECOVERY AFTER 22 HR IN AIR	AIR	62 HR IN R22 LIQUID	206 HR IN R22 LIQUID
Formvar	2.5–2.9	101	10	35	106	1	41	—	—	—
A	2.6–2.7	161	29	78	154	5	195	—	—	—
B	2.8	—	—	—	28	1	5	—	—	—
C	2.4–2.5	—	—	—	217	1	7	—	—	—
D	2.2–2.3	245	102	235	176	15	234	—	—	—
E	2.3–2.5	55	65	46	49	93	37	54	89	95
F	2.3–2.4	59	87	53	59	28	34	56	6	7
G	3.0–3.2	—	—	—	24	1	4	—	—	—
H	2.9	35	36	31	33	25	18	—	—	—
I	2.9–3.0	—	—	—	42	>1000	405	53	>1600	888
	1.7–1.2	21	39	18	—	—	—	—	—	—

Tables 1 and 2 show that formvar, polyacrylonitrile, medium functionality terephthalate polyesters, polyesteramide, and formvar-modified-epoxies soften considerably and lose almost all of their repeated scrape abrasion resistance after aging in Refrigerant 22 liquid for approximately 20 hr. The polyamide and the polyurethane modified polyesteramide appear not to be adversely affected in Refrigerant 22 liquid; in fact, the repeated scrape abrasion values were improved. The amine catalyzed epoxy is not affected in Refrigerant 22 gas but softens con-

TABLE 4....REPEATED SCRAPE ABRASION RESISTANCE OF VARIOUS TYPES OF WIRE ENAMELS IN AIR, REFRIGERANT 12 GAS AND LIQUID

(Room Temperature—450 Gram Load—0.016 in., Diam. Needle—Average of 4 Values in Cycles)

ENAMEL	BUILD (MILS)	AIR	AFTER 22 HR IN R12 GAS	AFTER AN ADDITIONAL 22 HR IN R12 LIQUID
Formvar	2.5–2.9	99	326	>588
A	2.6–2.7	67	>998	814
B	2.8	28	57	260
D	2.2–2.3	309	170	>1000
E	2.3–2.5	46	22	98
F	2.3–2.4	58	53	188
G	3.0–3.1	63	165	263
H	2.9	37	10	16
I	2.9–3.0	41	39	>1059

siderably in the liquid. The mechanism here is not fully understood at this time. Theoretically, the amount of Refrigerant 22 dissolved in the enamel should be the same under equilibrium conditions. Further testing to verify the results is planned for the near future.

Gaseous Refrigerant 12 behaves differently with film type insulations than Refrigerant 22. Instead of the enamels being softened, they become even more resistant to repeated scrape abrasion, probably due to a lubricating effect. Table 2 lists test values for the various wire enamels tested to date. As yet, these materials have not been tested in liquid Refrigerant 12.

Two other types of tests which are planned are those at elevated temperatures, and those in which compressor oils have been added to the refrigerants.

DISCUSSION

J. B. KELLEY, Columbus, Ohio: When the diameter of the needle was referred to as being 0.28, is that the point of the needle?

R. T. DIVERS, Syracuse, N. Y.: Was the vapor saturated when the enameled wire samples were in the vapor?

AUTHORS' CLOSURE (Mr. Trunzo): In reply to Mr. Kelley's question, the needle is transverse and on its side. It is unfortunate that this was not made clear.

Mr. Divers asked about vapor saturation. The samples were in the critical pressure region.

B. J. EISEMAN, JR., Wilmington, Del.: We have found that the halogenated hydrocarbons sometimes are retained very tenaciously by resins and plastics. This may be a factor in connection with these coated wires and could lead to a change in the coefficient of friction. The presence of retained refrigerant could be checked by a chloride analysis, after long exposure of the wire to air. Perhaps there would be a correlation between a better scrape test and the retention of refrigerant in the sample. This retention may occur not only with Refrigerant 12 but with any of the fluorinated hydrocarbon refrigerants and depends on the resin and the particular refrigerant. While it may not be relevant here, it is surprising how tightly the refrigerant is held in the resin.

MR. TRUNZO: It can be noticed from the values reported that there are differences in the scrape abrasion resistance for different materials.

MR. EISEMAN: This factor may not account for the spread, but it might account for the trend.

1695

No. 1695

DETERMINATION OF THE EFFECTIVENESS OF WINDOW SHADING MATERIALS ON THE REDUCTION OF SOLAR RADIATION HEAT GAIN

By R. C. JORDAN* AND J. L. THRELKELD**, MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, in cooperation with the Mechanical Engineering Department, University of Minnesota, Minneapolis, Minn.

WHEN solar radiation is transmitted through a glass window and a shading material is interposed between the room and the glass, a portion of this energy is transmitted through the shade directly, unless opaque, a part is reflected from the shade back to the glass and a part is absorbed by the shade itself. That portion of the energy absorbed by the shading material warms the shade to a temperature above that of the surrounding air. This, in turn, results in the transmission of heat from the shade, both to the interior of the room and in most cases back to and through the glass. Thus, a portion of the radiant energy incident upon the shade is added to the room load and a portion is retransmitted through the glass to the outside. These researches are concerned with the development of methods for the determination of the effectiveness of shading materials in reducing solar loads transmitted through and from shading materials into room interiors and the comparison of typical experimental results obtained by two test methods.

Two different procedures were used. *First*, the University of Minnesota solar calorimeters were adapted to testing shading materials and the proportion of the incident solar energy passing through the shading materials located in the calorimeters was measured. *Second*, determinations were made of the transmissivities

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and reflectivities of the shading materials independent of the calorimeter tests and equations developed by which such physical factors could be used to calculate the shade transmission factors. Both procedures are described and typical test results compared.

SOLAR CALORIMETER TESTS

The two solar calorimeters used in these studies have been described in detail in an earlier publication.[1] These solar collectors or calorimeters are identical in con-

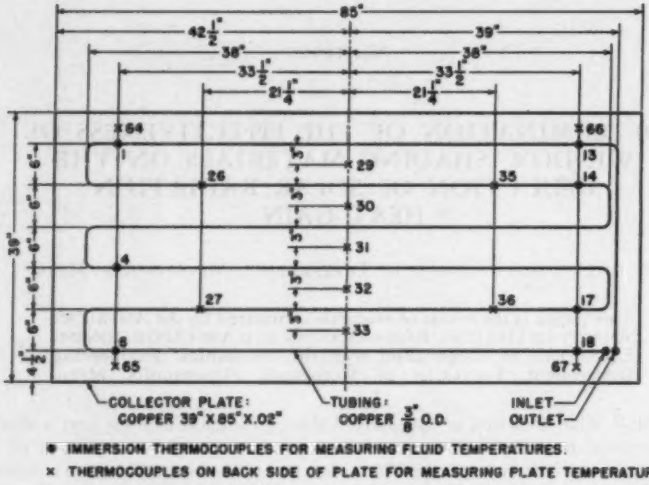

FIG. 1. . . . LOCATION OF PLATE AND FLUID TEMPERATURE MEASURING THERMOCOUPLES

struction and each is composed of a wooden enclosure, supporting frame, glass cover, and a copper collector plate with attached tubing for the absorption of heat loads. The calorimeter boxes are 94½ in. long, 48½ in. high, and 18 in. deep and are insulated with 4 in. of cork insulation on all sides excepting that containing the glass cover. The glass front is mounted on a steel frame and the glass opening itself is 39 in. × 85 in. The glass is ¼ in. water white plate glass with a total solar transmittance of 91 percent for normal incident solar radiation. The copper collecting plates are located 4½ in. behind the glass covers and are coated with a black absorbing surface having a spectral reflectance in the visible range of 0.0395. Heat passing through the glass covers and through any interposed shading materials is collected on the copper plates and hence carried away by the transfer fluid

[1] R. C. Jordan and J. L. Threlkeld: ASHAE RESEARCH REPORT No. 1583—Laboratory for solar energy study at Minnesota (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 461).

circulating in serpentine copper tubing soldered to the front side of the plates. The shading material under test was installed with the edges sealed between the glass cover and the copper plate at a distance of 1⅝ in. from the glass cover. Although the collectors are hinged to permit tilting to any position between horizontal and vertical, all tests were conducted with the collectors in the southfacing vertical position in order to simulate windows.

The collectors incorporated two separate fluid circulating systems, one for each collector. Each system consists of a storage tank for the circulating fluid, a pump,

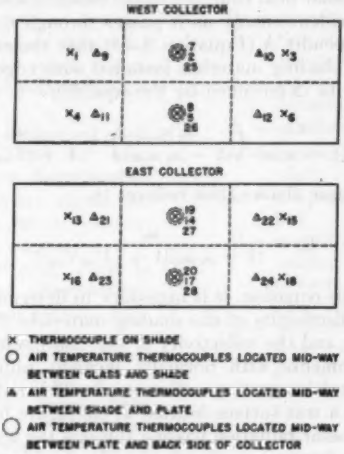
WEST COLLECTOR
EAST COLLECTOR
× THERMOCOUPLE ON SHADE
○ AIR TEMPERATURE THERMOCOUPLES LOCATED MID-WAY BETWEEN GLASS AND SHADE
△ AIR TEMPERATURE THERMOCOUPLES LOCATED MID-WAY BETWEEN SHADE AND PLATE
○ AIR TEMPERATURE THERMOCOUPLES LOCATED MID-WAY BETWEEN PLATE AND BACK SIDE OF COLLECTOR

FIG. 2. . . .LOCATION OF SOLAR COLLECTOR AIR TEMPERATURE MEASURING THERMOCOUPLES

a refrigeration unit for the absorption of the collected heat, a flow controller, and a flow-rate sensing element and a recorder. The fluid temperature in the storage tank was maintained at a constant value by a thermostatically controlled refrigeration system. In addition the flow controller maintained a constant outlet temperature by actuating a valve which varied the flow rate so as to maintain the outlet temperature at a fixed point. The flow-rate sensing element and recorder provided a continuous record of the flow rate throughout the tests. Since the amount of heat transmitted through the shading materials was small, the flow rate was also determined by actual weight measurement.

All temperatures were determined by copper-constantan thermocouples with the millivolt output measured by a self-balancing precision potentiometer. Four immersion thermocouples were used for measuring the inlet and outlet fluid temperatures of the calorimeters and 20 thermocouples for measuring the fluid and plate temperatures at various locations. In addition, 12 thermocouples were used

for measuring the temperature of the shading material itself and 16 thermocouples for measuring the air temperatures inside the collector box. The locations of these thermocouples are shown in Figs. 1 and 2.

The incident total solar radiation was measured by means of a 50-junction pyrheliometer mounted in the vertical position. Diffuse radiation measurements were made by shielding the element of the pyrheliometer with a shading ball.

DETERMINATION OF SHADE TRANSMISSION FACTORS FROM PHYSICAL CONSTANTS

The experimental shade transmission factor, E'_s, determined by the calorimeter tests is the ratio of the solar heat collected to the incident solar radiation, corrected for the reduction in incident energy as it passes through the cover glass. However, it is shown in Appendix A (Equation A-15) that the calculated shade transmission factor, E_s, for shading materials installed with edges sealed in such flat-plate calorimeters may be determined by the equation

$$E_s = \frac{1}{(1 - \rho_s \rho_G)} \left(\frac{\alpha_p \tau_s}{(1 - \rho_p \rho_{sB})} \right) + \frac{\alpha_s}{1 + (U_{so}/C_{sp})} \quad . \quad . \quad . \quad . \quad (1)$$

and, in the case of opaque shades, this reduces to

$$E_s = \frac{\alpha_s}{(1 - \rho_s \rho_G)[1 + (U_{so}/C_{sp})]} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In order to evaluate this equation, it is necessary to determine experimentally the transmissivities and reflectivities of the shading materials, the absorptivity of the blackened copper plate and the reflectivity of the calorimeter cover glass.

Reflectivity measurements with normally incident radiation were made by means of a hemispherical integrating radiometer.[2] This radiometer was equipped with a thermopile and a test surface located at conjugate foci on the diameter of the hemisphere. The solar radiation passing through the aperture of the radiometer was first allowed to impinge on the thermopile as an indication of the incident solar radiation. Next, solar energy was reflected from the test surface onto the thermopile with the necessary integration of energy reflected at all angles accomplished by the hemispherical surface.

The ratio of the energy reflected to the thermopile from the test surface to that directly incident onto the thermopile provides a measure of the relative reflectivity of the test surface when compared to a roughened surface with known absolute reflectivity. The reference used was a magnesium oxide surface for which the reflectivity in the solar spectrum is approximately 0.95.

Additional reflectivity measurement checks were made with an integrating sphere radiometer in which the inside of the sphere was coated with magnesium oxide so that radiation reflected from the surface under test provided a uniform radiation field throughout the sphere and measurement of this and comparison with a surface of known reflectivity provides a measurement of the relative reflectivity.

Transmissivity measurements of the cover glass used on the calorimeters, the shading materials, and combinations of shading materials and glass were measured by means of a 9- × 18- × 18-in. chamber placed over a 50-junction pyrheliometer. The front of this chamber was open to permit the insertion of a section of glass

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[2] R. C. Birkebak and J. P. Hartnett: Measurements of the total absorptivity for solar radiation of several engineering materials (*ASME Transactions*, February 1958, p. 373).

NOMENCLATURE

A_s = area of shade, square feet.

C_{sG} = thermal conductance from outside surface of shading material to inside surface of cover glass or window, Btu per (hour) (square foot) (Fahrenheit degree).

C_{sp} = thermal conductance from inside surface of shading material to collector plate, Btu per (hour) (square foot) (Fahrenheit degree).

E_s = shade transmission factor for solar calorimeter tests as calculated by individual coefficients, dimensionless.

E'_s = experimentally determined shade transmission factor for solar calorimeter tests, dimensionless.

e_s = shade transmission factor for shade applied to a window, dimensionless.

F_s = ratio of area of sunlit shade to total shade area, dimensionless.

f_i = coefficient of heat transfer from inside surface of shade to room, Btu per (hour) (square foot) (Fahrenheit degree).

f_o = coefficient of heat transfer from outside surface of cover glass or window to outside environment, Btu per (hour) (square foot) (Fahrenheit degree).

I = rate of incidence of direct and diffuse solar radiation upon outside surface of cover glass or window, Btu per (hour) (square foot).

k_G = thermal conductivity of cover glass or window, Btu per (hour) (square foot) (Fahrenheit degree per inch).

Q_A = total energy transferred through shade absorbed by collector plate or room, Btu per hour.

Q_1 = rate of incidence of solar radiation upon outside surface of shade, Btu per hour.

Q_2 = solar radiation directly transmitted through shade, Btu per hour.

Q_3 = solar radiation absorbed by outside surface of shade, Btu per hour.

Q_4 = energy transferred by normal transmission from inside surface of shade to collector plate or room, Btu per hour.

Q_5 = energy transferred by normal transmission from outside surface of shade through cover glass or window to outside, Btu per hour.

Q_6 = directly transmitted solar radiation absorbed by collector plate, Btu per hour.

t_i = interior temperature of room, Fahrenheit.

t_o = outside temperature, Fahrenheit.

t_p = temperature of the collector plate, Fahrenheit.

t_s = temperature of shade, Fahrenheit.

U = overall coefficient of heat transmission from outside to interior of room, Btu per (hour) (square foot) (Fahrenheit degree).

U_{so} = coefficient of heat transmission from outside surface of shade through cover glass or window to outside, Btu per (hour) (square foot) (Fahrenheit degree).

x_G = thickness of cover glass or window, inches.

α_p = absorptivity of collector plate for solar radiation, dimensionless.

α_s = absorptivity of outside surface of shade for solar radiation, dimensionless.

ρ_G = reflectivity of glass surface for solar radiation, dimensionless.

ρ_p = reflectivity of collector plate for solar radiation, dimensionless.

ρ_s = reflectivity of outside surface of shade for solar radiation, dimensionless.

$\rho_{s,B}$ = reflectivity of inside surface of shade for solar radiation, dimensionless.

τ_G = transmissivity of cover glass or window glass for solar radiation, dimensionless.

τ_s = transmissivity of shade for solar radiation, dimensionless.

similar to that used on the collectors and panels of the shading materials. This permitted measurements of the incident solar radiation without the interposed glass or shading material and of the incident solar radiation transmitted through the shade and glass. Determinations of the absorptivity factors, α_s, were made by the equation $\alpha_s = 1 - \tau_s - \rho_s$, once the transmissivities and reflectivities were known.

TEST RESULTS

In these studies five different types of shading materials were used and each may be briefly described as follows:

Shade Material A: Bleached cotton cloth, vinyl resin solution coating, and fireproofed, partially translucent, slightly glossy, buff-colored surfaces.

Shade Material B: Bleached cotton cloth, vinyl resin solution coating, partially translucent, white in color, low surface gloss.

Shade Material C: Bleached kraft paper base, vinyl resin solution coating, partially translucent, embossed white surfaces on both sides.

Shade Material D: Cloth base, treated and coated with vinyl resin, opaque, slightly glossy white finish on both sides of the material.

Shade Material E: Vinyl-coated cotton base, opaque, laminated with embossed aluminum foil on one side and white, coated surface on the other.

It was found from reflectivity measurements that the two sides of materials A, C, and D had different reflectivities. These materials were fitted in the solar calorimeter with the dull side (the side with lower reflectivity) facing out, according to the manufacturer's instructions. Material E was tested with the embossed aluminum foil facing out. These 5 shading materials were tested in the solar calorimeters by sealing the material under the back of the glass cover with an air space of 1⅝ in. between the cover and the shade.

This provided an idealized condition in which no air movement could occur around the edges of the shading material and differs from practice unless channels or similar devices are provided in which the shade may be raised or lowered. The testing and evaluation are comparable to that for shading materials in a fully drawn position with no attempt to simulate partially drawn shades.

During actual testing, one calorimeter was equipped with shading material B as a standard for comparison since this material appeared to have a comparatively high transmission factor and therefore allowed better accuracy in determining heat collected. The alternate shading material was installed in the second collector. An additional test was made with shading material D compared directly with shading material E since these two both appeared to have quite low shade transmission factors under identical solar conditions.

Fig. 3 shows typical curves obtained for the percentage of incident solar heat collected on a cloudless day by the calorimeters when equipped with different shading materials. The percentage of heat collected was comparatively constant through the hours near solar noon but dropped rapidly during the late hours and rose rapidly during the earlier morning hours when the angles of incidence between the sun and the calorimeters were comparatively high.

Since the experimental shade transmission factor, E'_s, as determined by the calorimeter test is inherently a measurement made under controlled conditions

similar to actual use and does not necessitate the determination of the physical constants required for evaluation of E_s by Equation 1, the calorimeter test procedure was regarded as yielding the more accurate results. Both the reflectivity and transmissivity apparatus and measurement procedures require further re-

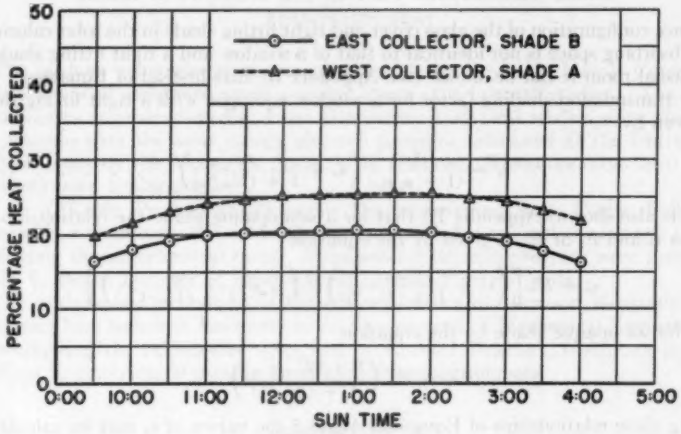

FIG. 3. . . . TYPICAL SOLAR CALORIMETER TEST RESULTS COMPARING TWO SHADING MATERIALS

finement before equivalent confidence can be placed in the calculated shade transmission factor, since the equation to determine E_s is quite sensitive to small variations in these measurements.

Table 1 shows a comparison of the calorimeter test results and the calculated shade transmission factors for all shading materials tested. The sensitivity of the results obtained in determining the calculated shade transmission factors by use of Equation 1 may be illustrated by analyzing the calculations for shade E. The average of the experimentally determined values of the reflectivity, ρ_s, was 0.85

TABLE 1. . . . COMPARISON OF EXPERIMENTAL AND CALCULATED SHADE TRANSMISSION FACTORS FOR SOLAR CALORIMETER

(*Shade edges sealed*)

SHADE MATERIAL	EXPERIMENTAL SHADE TRANSMISSION FACTOR, E'_s	CALCULATED SHADE TRANSMISSION FACTOR, E_s
A	0.28	0.30
B	0.23	0.26
C	0.23	0.26
D	0.18	0.18
E	0.14	0.14

and this value provided an exact check between the experimental and the calculated shade transmission factors, $E'_s = E_s = 0.14$. Yet the range of values for ρ_s from which this average was obtained permitted a range of values of E_s between 0.17 and 0.11.

SHADE TRANSMISSION FACTORS FOR WINDOWS IN ROOMS

Since configuration of the glass cover and tight fitting shade in the solar calorimeter absorbing space is not identical to that of a window and a tight fitting shade in the usual room it can be shown (see Appendix B) that instead of Equation 1 the solar transmission shading factor for a window equipped with a tight fitting shade is given by

$$\epsilon_s = \frac{1}{(1 - \rho_s \rho_G)} \left(\tau_s + \frac{\alpha_s}{1 + (U_{so}/f_i)} \right) \quad (3)$$

It is also shown (Appendix B) that for a non-opaque shade the relationship between ϵ_s and E_s or E'_s is given by the equation

$$\epsilon_s = E'_s \left[\left(\tau_s + \frac{\alpha_s}{1 + (U_{so}/f_i)} \right) \Big/ \left(\tau_s + \frac{\alpha_s}{1 + (U_{so}/C_{sp})} \right) \right] \quad (4)$$

and for an opaque shade by the equation

$$\epsilon_s = E'_s \left(\frac{1 + (U_{so}/C_{sp})}{1 + (U_{so}/f_i)} \right) \quad (5)$$

Using these relationships of Equations 4 and 5 the values of ϵ_s may be calculated. Table 2 shows a summary of calculations for the shade transmission factor ϵ_s for windows for angles of incidence between 0 and approximately 60 deg.

TABLE 2. . . .SUMMARY OF FACTORS INVOLVED IN CALCULATION OF SHADE TRANSMISSION FACTORS FOR WINDOWS LOCATED IN ROOMS
(*Shade edges sealed*)

SHADE MATERIAL	E'_s	τ_s	α_s	U_{so}	U_{so}/C_{sp}	U_{so}/f_i	ϵ_s
A	0.28	0.135	0.250	0.86	0.716	0.589	0.29
B	0.23	0.147	0.143	0.86	0.716	0.589	0.24
C	0.23	0.125	0.185	0.86	0.716	0.589	0.24
D	0.18	0.0	0.27	0.86	0.716	0.589	0.19
E	0.14	0.0	0.155	0.32	0.267	0.219	0.15

SUMMARY

Window shading material transmission factors for 5 different materials were determined both by solar calorimeter tests, and by calculation through independently measured reflectivities and transmissivities of the materials. Equations have been derived for calculation of the transmission factors, both in application to a calorimeter and also in application to a window and room. The checks between the experimental and calculated shade transmission factors are good in all cases.

However, the sensitivity of the calculated transmission factors to determination of the reflectivity and transmissivity is such that experimental variations in these

values materially affect the overall transmission values. In those cases where the physical constants may be reliably evaluated the transmission factors may be calculated accurately, but in those cases where these values may be questioned, calorimeter tests appear necessary.

ACKNOWLEDGMENT

The authors acknowledge the cooperation of the Reynolds Metals Company, whose financial assistance made possible these experimental studies. The solar calorimeters were developed and the calorimeter analytical studies were made in conjunction with the solar energy research program sponsored at the University of Minnesota by the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS.

J. F. O'Keefe, Product Development Department and A. J. Castle, Foil Development Laboratory, Reynolds Metals Co., provided valuable assistance in coordinating the experimental tests. Materials for the test program were provided by B. E. Beale, Director of Development, Joanna Western Mills.

The authors also acknowledge the able assistance of Harrison Benjamin and Benjamin Liu, Research Assistants in the Department of Mechanical Engineering, in conducting the calorimeter tests, and of Richard Birkebak, Instructor in Mechanical Engineering, in making the reflectivity measurements.

APPENDIX A

DETERMINATION OF SHADE TRANSMISSION FACTOR FOR SHADED COLLECTOR PLATE WITH PLATE TEMPERATURE EQUAL TO OUTDOOR TEMPERATURE

Fig. A-1 shows schematically a flat plate collector with a shading material interposed between the collector plate and the cover glass. In this analysis it is assumed that the collector plate is maintained at a temperature equal to the outdoor ambient temperature.

The solar radiation transmitted through the glass cover is

$$Q_1 = F_s A_s \tau_G I \quad \text{(A-1)}$$

A part of this energy may be transmitted through the shade directly, a part reflected and a part absorbed by the shade. Part of the reflected energy is reflected back again to the shade by the glass cover and this tends to increase the radiation transmitted and absorbed. The total radiation directly transmitted through the shade is

$$Q_2 = \tau_s Q_1 (1 + \rho_s \rho_G + \rho_s^2 \rho_G^2 + \ldots) = \frac{\tau_s Q_1}{1 - \rho_s \rho_G} \quad \text{(A-2)}$$

The total radiation absorbed by the shade is

$$Q_3 = \frac{\alpha_s Q_1}{1 - \rho_s \rho_G} \quad \text{(A-3)}$$

The radiation absorbed by the shade will cause an increase in shade temperature above the plate temperature. If the shade has negligible temperature drop and is at a uniform temperature, t_s, part of the energy absorbed will be transferred outwardly through the glass, or

$$Q_3 = Q_4 + Q_5 \quad \text{(A-4)}$$

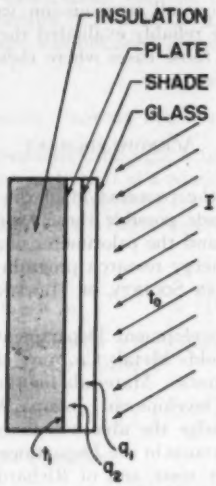

FIG. A-1. . . .SCHEMATIC SOLAR CALORIMETER WITH SHADE

or

$$Q_3 = C_{sp} A_s (t_s - t_p) + U_{so} A_s (t_s - t_o) \quad \text{(A-5)}$$

where

$$U_{so} = \frac{1}{\frac{1}{C_{sG}} + \frac{x_G}{k_G} + \frac{1}{f_o}} \quad \text{(A-6)}$$

Since in the tests, the collector plate temperature was maintained equal to the outdoor air temperature, Equation A-5 may be written as

$$Q_3 = A_s (t_s - t_o) \quad (C_{sp} + U_{so}) \quad \text{(A-7)}$$

By Equations A-1, A-3, and A-7

$$t_s - t_o = \frac{\alpha_s F_s \tau_G I}{(1 - \rho_s \rho_G)(C_{sp} + U_{so})} \quad \text{(A-8)}$$

By Equations A-1, A-3, A-4, A-5, and A-8

$$Q_4 = \frac{\alpha_s F_s \tau_G}{1 - \rho_s \rho_G} I \left(1 - \frac{U_{so}}{C_{sp} + U_{so}}\right) A_s \quad \text{(A-9)}$$

The directly transmitted radiation which is absorbed by the collector plate is

$$Q_6 = \frac{\alpha_p}{(1 - \rho_p \rho_{s,B})} Q_2 \quad \text{(A-10)}$$

and by Equations A-1, A-2, and A-10

$$Q_6 = \frac{\alpha_p \tau_s F_s \tau_G I A_s}{(1 - \rho_p \rho_{s,B})(1 - \rho_s \rho_G)} \quad \text{(A-11)}$$

The total energy absorbed by the plate is

$$Q_A = Q_4 + Q_6 \quad \text{(A-12)}$$

and by Equations A-9, A-11, and A-12

$$Q_A = \frac{F_s A_s \tau_G I}{(1 - \rho_s \rho_G)} \left(\frac{\alpha_p \tau_s}{(1 - \rho_p \rho_{s,B})} + \frac{\alpha_s}{1 + (U_{so}/C_{sp})} \right) \quad \text{(A-13)}$$

If the shade transmission factor E_s is defined as

$$E_s = \frac{Q_A}{Q_I} \quad \text{(A-14)}$$

Then by Equations A-1, A-13, and A-14

$$E_s = \frac{1}{(1 - \rho_s \rho_G)} \left(\frac{\alpha_p \tau_s}{(1 - \rho_p \rho_{s,B})} + \frac{\alpha_s}{1 + (U_{so}/C_{sp})} \right) \quad \text{(A-15)}$$

APPENDIX B

DETERMINATION OF SHADE TRANSMISSION FACTOR FOR A SHADED WINDOW

Fig. B-1 shows schematically a shaded window in relation to a comparatively large room. By means analogous to those of Appendix A, is the total energy transferred to the room is

$$Q_A = Q_2 + Q_4 \quad \text{(B-1)}$$

where

$$Q_2 = \frac{\tau_s F_s A_s \tau_G I}{(1 - \rho_s \rho_G)} \quad \text{(B-2)}$$

$$Q_4 = f_i A_s (t_s - t_i) \quad \text{(B-3)}$$

and

$$Q_4 = \frac{\alpha_s F_s A_s \tau_G I}{(1 - \rho_s \rho_G)} - U_{so} A_s (t_s - t_o) \quad \text{(B-4)}$$

By Equations B-3 and B-4

$$t_s = \frac{\dfrac{\alpha_s F_s \tau_G I}{(1 - \rho_s \rho_G)} + f_i t_i + U_{so} t_o}{f_i + U_{so}} \quad \text{(B-5)}$$

By Equations B-3 and B-5

$$Q_4 = \frac{\alpha_s F_s A_s \tau_G I}{(1 - \rho_s \rho_G)(1 + (U_{so}/f_i))} + U A_s (t_o - t_i) \quad \text{(B-6)}$$

where

$$U = \frac{1}{(1/U_{so}) + (1/f_i)} \quad \ldots \quad \text{(B-7)}$$

Thus by Equations B-1, B-2, and B-6

$$Q_A = \frac{F_s A_s \tau_G I}{(1 - \rho_s \rho_G)} \left(\tau_s + \frac{\alpha_s}{1 + (U_{so}/f_i)} \right) + U A_s (t_o - t_i) \quad \ldots \quad \text{(B-8)}$$

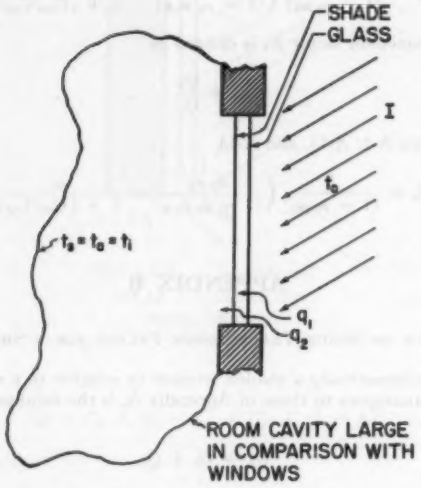

FIG. B-1. . . .SHADED WINDOW IN RELATION TO ROOM

If the shade transmission factor is defined as

$$e_s = \frac{Q_A}{Q_I} = \frac{Q_A}{F_s A_s \tau_G I} \quad \ldots \quad \text{(B-9)}$$

Then by Equations B-8 and B-9

$$e_s = \frac{1}{(1 - \rho_s \rho_G)} \left(\tau_s + \frac{\alpha_s}{1 + (U_{so}/f_i)} \right) + \frac{U (t_o - t_i)}{F_s \tau_G I} \quad \ldots \quad \text{(B-10)}$$

Equation B-10 is general and expresses the performance of the shade both with respect to solar transmission and to normal heat transmission. For solar effects only, or with $t_o = t_i$, Equation B-10 reduces to

$$e_s = \frac{1}{(1 - \rho_s \rho_G)} \left(\tau_s + \frac{\alpha_s}{1 + (U_{so}/f_i)} \right) \quad \ldots \quad \text{(B-11)}$$

In Equation A-15 of Appendix A, the term $\alpha_p/(1 - \rho_p\rho_{s,B})$ was essentially unity for all shading materials studied. With this approximation one may write by Equations A-15 and B-11

$$\epsilon_s = E_o \frac{\tau_s + \dfrac{\alpha_s}{1 + (U_{so}/f_i)}}{\tau_s + \dfrac{\alpha_s}{1 + (U_{so}/C_{sp})}} \quad \ldots\ldots\ldots \text{(B-12)}$$

DISCUSSION

L. F. SCHUTRUM, Cleveland, Ohio (WRITTEN): The effectiveness of window shade material on the reduction of solar gain is reported in this paper for the condition in which the edges of the shade are sealed to the frame and thus no air circulates between the room and the space between the shade and the glass. The next paper* to be presented entitled *Heat Flow Through Glass with Roller Shades*, gives the effectiveness of various roller shades in reducing the heat gains from solar radiation; but with the roller shades mounted in a conventional manner so that air circulates between the space and the room.

The result of these two investigations can be compared by computation from the properties of the shade materials given in Table 2 of this paper, Equation B-7 of Chapter 1696 for the convection and radiation component of heat gain, and the transmitted component which is readily calculated. If there were air circulation, the shade transmission factors for shades A through D would be about 25 percent higher on an average than those tabulated in Table 2. These shades all have high emissivity surfaces. For shade E, which has a low emissivity surface on the glass side, the transmission factor with air circulation would be about 12 percent higher than for the sealed condition. A value of 3 Btu/(hr)(sq ft)(F deg) for the overall outdoor coefficient was used in the calculation of the transmission factor for roller shades; but the authors do not state the values used in their calculations. However, a change in the combined outdoor coefficient of 3 to 4 would not appreciably alter the transmission factors.

The transmission factors for sealed shades are about 10 to 30 percent lower than for unsealed shades, depending upon the emissivity for long wave radiation of the shade.

The authors made all of their experiments with a window glass having a low solar absorptance and have not included the solar energy absorbed by the glass in their energy balance equations. The energy balance on the shade is given in the Appendix by Equation A-5.

$$Q_3 = C_{sp}A_s(t_s - t_p) + U_{so}A_s(t_s - t_o) \quad \ldots\ldots\ldots \text{(A-5)}$$

when it is assumed that the solar absorption by the glass is small and can be neglected. For regular plate, heat absorbing glasses, the glass absorptance term can be an important factor and should be included in Equation A-5. This equation would then be as follows:

$$Q_3 = C_{sp}A_s(t_s - t_p) + C_{sG}A_s(t_s - t_G) \quad \ldots\ldots\ldots \text{(A)}$$

and the temperature of the glass (t_G) can be obtained from the energy balance of the glass per unit area.

$$\alpha I_G = C_{sG}(t_G - t_s) + f_o(t_G - t_o) \quad \ldots\ldots\ldots \text{(B)}$$

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* *Editor's Note*: The reference is to the Chapter 1696 beginning on page 697 of this volume.

Where αI_g is the solar energy absorbed by the glass, and the temperature differential across the glass surfaces are neglected, the glass temperature becomes

$$t_G = \frac{\alpha I_G + C_{sG} t_s + f_o t'_o}{C_{sG} + f_o} \quad \ldots \ldots \ldots \quad (C)$$

Substituting C in A gives the energy balance on the shade including the energy absorbed by the glass.

$$Q_3 = C_{ps} A_s (t_s - t_p) + C_{sG} A_s \left[t_s - \frac{\alpha I_G + C_{sG} t_s + f_o t'_o}{C_{sG} + f_o} \right] \quad \ldots \ldots \quad (D)$$

This Equation, D, will reduce to Equation A-5 of the paper, when the solar energy absorption by the glass is zero.

The final Equations A-15 and B-11 of the paper, for a more general application, should include an additional term for the solar energy absorbed by the glass. The term to be added is:

$$\frac{U \alpha_G}{f_o \tau_G} \left[\frac{1}{F_s} + \frac{\rho_s \tau_G}{1 - \rho_s \rho_G} \right]$$

G. R. MUNGER, Toledo, Ohio: Would the authors expect these same shade factor values for orientations other than South?

R. W. MCKINLEY, Pittsburgh, Pa.: Can the author indicate whether the report ought to state clearly the type of glazing medium with which the shade material is associated? It would seem reasonable to believe that the reflective characteristics of the combination would become increasingly or decreasingly dependent upon the transmittance characteristics of the media; *i.e.*, the medium might be moderately absorptive or it might have a very high transmittance for the total energy spectrum. From the standpoint of applying shade factors of this type, it would seem important to know that there is an effect that is relative to the type of media with which the factors are associated.

AUTHORS' CLOSURE (Mr. Threlkeld): The question by Mr. Munger is whether the shade transmission factors would be affected by orientation. The tests reported on were carried out with the collectors in a south-facing vertical position. The shade transmission factors should be primarily a function of angle incidence only. Therefore, the results should be applicable to other orientations.

In answer to Mr. McKinley's remarks I should point out that the results of this paper apply to ordinary house glass windows where the shades are tightly sealed and where solar radiation absorption by window glass may be neglected.

The authors wish to thank Mr. Schutrum for relating these two papers (1695 and 1696) and for extending the information in the authors' paper. He has presented an estimate of the difference in shade transmission factors between the sealed shade and the unsealed shade.

Mr. Schutrum asked what value of outdoor surface coefficient was used in the calculations. The value used was 4.0 Btu per (hr)(sq ft)(F deg). With regard to his other comments it is necessary to apply the correction shown by Mr. Schutrum for window glass where heat absorption may not be neglected. For ordinary house window glass this correction is not necessary.

1696

No. 1696

HEAT FLOW THROUGH GLASS WITH ROLLER-SHADES

By NECATI OZISIK* AND L. F. SCHUTRUM**, CLEVELAND, OHIO

This paper is the result of research carried out by the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio

THIS investigation of heat flow through glass and roller-shade combinations under summer and winter conditions was under the guidance of the former ASHAE Technical Advisory Committee on Heat Transfer Through Fenestration.† The type and color of shade materials to be tested were selected at a meeting attended by representatives of the shade material manufacturers and members of the ASHAE Research Laboratory staff. The color and material of the shade and its positioning with respect to the glass were recognized as significant factors influencing the heat flow. Roller-shade materials used for testing included:

1. Both sides white, vinyl plastic coated fabric, opaque;
2. Both sides white, vinyl plastic coated fabric, translucent;
3. Both sides dark green, vinyl plastic coated fabric, opaque;
4. One side aluminum pigmented coated, other side white vinyl plastic coated fabric, opaque;
5. One side dark green vinyl plastic coated fabric, other side aluminum foil (*i.e.* aluminum foil was pasted on one side of the dark, green material), opaque; and
6. Both sides aluminum foil, metallic, opaque.

Opaque materials did not permit any solar radiation to pass through them, whereas, some solar radiation passed through the translucent material.

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* Research Engineer, ASHRAE Research Laboratory.

** Research Supervisor, ASHRAE Research Laboratory. Member of ASHRAE.

† *Personnel:* D. D'Eustachio, *chairman*; R. C. Jordan, *vice chairman*; T. C. Carson, O. D. Englehart, W. B. Ewing, C. W. Kelly, J. B. Leavy, E. C. Miles, Bruno Morabito, F. W. Mowrey, G. R. Munger, O. L. Pierson, D. J. Vild, O. F. Wenzler, C. J. Youngblood.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N. Y., June 1959.

Special tests were made at the Laboratory to determine the properties of these materials. The solar reflectance was determined by comparing reflectance of the shade with that of a magnesium carbonate block, using a pyrheliometer. Transmittance for solar radiation of the translucent material was also obtained with a pyrheliometer.

The emissivity of the materials for long-wave radiation was measured using an apparatus as described in an earlier report[1].

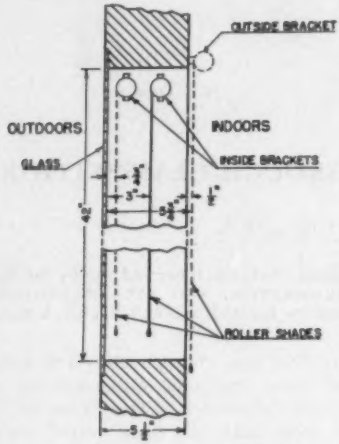

FIG. 1. . . .POSITIONS OF THE ROLLER-SHADES

The glasses used included ¼-in. regular plate glass and ¼-in. heat absorbing glass. The properties of the glasses and shade materials tested are given in Table 1. Properties of similar shade materials from different manufacturers were within 10 percent of the values given in this table.

The shade was positioned to the calorimeter window with usual brackets in the conventional manner. Spacing between the glass and shade was 5¾ in. with outside type bracket and ¾ or 3 in. with inside brackets (see Fig. 1).

TEST APPARATUS

Except for minor changes the solar calorimeter used in these studies was essentially as described in earlier reports[2, 3]. The heat gain through the 42-in. square test window was measured with the inside of the calorimeter maintained at a nominal temperature of 75 F.

Measurements included direct and diffuse components of the incident solar radiation, shade and glass temperatures, outdoor air temperature, air temperature in the

[1] Exponent numerals refer to References.

space between the shade and glass, and the usual measurements for heat flow determination.

The solar calorimeter is shown in Fig. 2.

HEAT FLOW THROUGH A GLASS AND ROLLER-SHADE COMBINATION AND THE EXPERIMENTAL RESULTS

Consider a single glass and roller-shade combination with solar radiation falling upon the glass surface. Part of this radiation is reflected, part is absorbed by the glass and shade, and, if the shade material is translucent, part is transmitted through

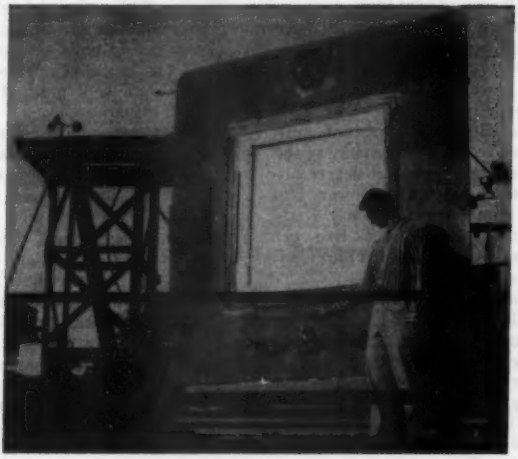

FIG. 2. . . .VIEW OF THE SOLAR CALORIMETER

the glass and shade into the room. The amount of energy transmitted can be calculated with reasonable accuracy knowing the transmittances of the glass and shade and the intensity of the solar radiation. The convection-radiation component of the heat gain into the room resulting from the solar energy absorption by the glass and shade, however, is less susceptible of mathematical prediction. The solar energy absorbed by the glass and roller-shade combination increases their temperatures until a balance is reached between the rate of heat absorption by the combination and the rate of heat dissipation from the combination by convection-radiation both into the room and to the outdoors.

By writing the energy balance equations for the roller-shade and glass, as given in Appendix A, it can be shown that, for a given glass and roller-shade combination and outside conductance, the convection-radiation component of the heat gain resulting from the solar heat absorption, *i.e.*,

$$CR - U(t_o - t_i),$$

is a function of the solar energy absorbed by the glass and roller-shade combination, *i.e.* αI_{gs}, and, with no solar radiation both terms are zero, *i.e.*

$$\alpha I_{gs} = O;\ CR = U(t_o - t_i)$$

This indicates that, for a given glass and roller-shade combination and outside conductance, if $CR - U(t_o - t_i)$ is plotted against αI_{gs}, the curve passes through the origin. During the tests, however, the outside conductance varied. In order

NOMENCLATURE

C = convection heat exchange, Btu per (hour) (square foot).
R = radiation heat exchange, Btu per (hour) (square foot).
CR = convection and radiation heat gain into the room, Btu per (hour) (square foot).
E = Stefan-Boltzman constant multiplied by the temperature of the surface in Fahrenheit degree absolute to the fourth power, Btu per (hour) (square foot).
h = thermal conductance Btu per (hour) (square foot) (Fahrenheit degree).
I_v = intensity of solar radiation on a vertical surface, Btu per (hour) (square foot).
K = a factor, dimensionless.
αI_g = solar energy absorbed by the glass, Btu per (hour) (square foot).
αI_s = solar energy absorbed by the shade, Btu per (hour) (square foot).
αI_{gs} = solar energy absorbed by glass-shade combination, Btu per (hour) (square foot).
t = temperature, Fahrenheit.
t'_g = temperature of glass with no solar radiation, Fahrenheit.
Δt = rise in glass temperature due to solar radiation, Fahrenheit.
Q = total heat gain, Btu per (hour) (square foot).
r = a factor, dimensionless.
S = a factor for determining the roller-shade temperature, Btu per (hour) (square foot) (Fahrenheit degree).
T = transmittance, dimensionless.
U = overall coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree).
U' = solar heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
α = absorptance, dimensionless.
ξ = emissivity for long-wave radiation, dimensionless.
γ = reflectance, dimensionless.

Subscripts

a, g, s = space between glass and shade, glass, and shade.
i, o = inside and outside.

to correlate the test data with the foregoing plot, the data were adjusted to the same outside conductance as explained in Appendix B.

For a 75 F indoor temperature, the $CR - U(t_o - 75)$ values from the test data, after being adjusted for an outside conductance of 3* Btu per (hr) (sq ft) (F deg) were plotted against αI_{gs}, in Fig. 3. As seen from this figure, the data for a given glass and roller-shade combination correlated with a straight line passing through the origin, and the slope of the line represented the fraction of the solar energy absorbed by the glass and roller-shade combination entering the room as convection-radiation heat gain. Knowing αI_{gs}, U and the slope of the line, the convection-

* 3 Btu per (hr) (sq ft) (F deg) was chosen because experimental data were closer to 3 than the commonly used value of 4.

radiation component of the heat gain can be calculated. If the shade material is translucent there is a transmitted component, which can be calculated from the known values of the transmittances of the glass and shade material and the intensity of the solar radiation.

For practical purposes, however, it was not so easy to calculate the heat gains in a manner just described. In order to simplify the calculations, the convection-radiation and transmitted components were combined into a single multiplier as

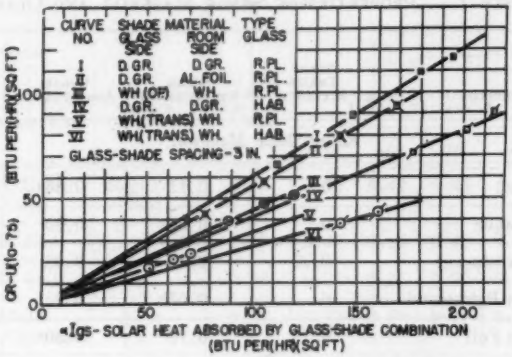

FIG. 3. . . .CONVECTION AND RADIATION HEAT GAIN FOR GLASS AND ROLLER-SHADE COMBINATIONS DUE TO SOLAR RADIATION

explained in Appendix C, and the total heat gain per unit area of the roller-shade and glass combination was reduced to Equation 1, thus:

$$Q = KI_v + U(t_o - 75) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

K = a factor depending on whether the solar radiation is diffuse or direct, the incidence angle of the direct beam with the glass surface, and the type of glass and roller-shade combination.
I_v = intensity of the solar radiation falling upon the glass surface.
U = overall coefficient of heat transfer.

The K and U values to be used in Equation 1 are tabulated in Table 2 for the glass and roller-shade combinations tested. K values were calculated as described in Appendix C using the values from Table 1 and Fig. 3, and adjusted for an inside convection heat transfer as suggested in Reference 4. The U values in Table 2 are based on the winter tests with an average outdoor air temperature of about 25 F and without solar radiation. However, the U values with higher outdoor temperatures in summer are not appreciably different from the U values given in this table.

The data in Table 2 are for a fully drawn roller shade, with 3-in. glass-to-shade spacing, and with edge openings of about ½-in. all around the shade.

Therefore, by taking the appropriate K and U values from Table 2, heat flow through a glass and roller-shade combination can be calculated from Equation 1, for summer and winter, with and without solar radiation.

The variation of glass to shade spacing from ¾ in. to 3 in. did not change the heat gain with solar radiation; with 5¾-in. spacing, however, the heat gains were about 5 to 10 percent higher. With no solar radiation, heat flow was practically unaffected by the variation of glass to shade spacing from ¾ in. to 5¾ in.

TABLE 1....PROPERTIES OF SHADE MATERIAL AND GLASS

MATERIAL	SOLAR TRANSMITTANCE	SOLAR REFLECTANCE	SOLAR ABSORPTANCE	EMISSIVITY FOR LONG-WAVE RADIATION
Roller Shade Material:				
Dark Green	0	0.12	0.88[a]	0.90
White Opaque	0	0.80	0.20	0.87
White Translucent	0.22	0.70	0.08	0.89
Aluminum Pigmented	0	0.67[b]	0.33	0.55
Aluminum Foil	0	0.70	0.30[c]	0.07
Type of Glass:				
Ordinary Window[d]	0.87	0.08	0.05	
Regular Plate	0.77	0.07	0.16	
Heat Absorbing	0.41	0.06	0.53	

[a] For design data 0.85 was used.
[b] At 70 degree incidence angle reflectance—was 0.40.
[c] Reference 6 gives 0.19 for a bright aluminum surface. Compared with bright aluminum, experiments indicated an absorptance of about 0.30 for the dull surface aluminum foil.
[d] No test made.
Note: All values are for normal incidence.

A few tests were made with the edges of the shade sealed to the window frame, with the shade half drawn, and with a fully open shade, and the overall coefficients of heat transfer for these conditions without solar radiation are included in Table 3.

Using values from Table 2 and Equation 1, the summer heat gains through different glass and roller-shade combinations for a 75 F indoor temperature were calculated for east, south and west orientations and for various hours of the day, and the data are presented in Table 4. For these calculations the outdoor air temperatures and the solar radiation intensities were taken as those given in the ASHAE GUIDE for a clear atmosphere, and at 40 deg North Latitude on August 1. It is to be noted that, in preparing Table 4, no allowance was made for the window setback. Therefore, this table gives only the heat gains through the sunlit section of the glass and roller-shade combination. The heat gains through the section in shadow were also calculated and are given in Table 5. In preparing Table 5, the outdoor con-

ditions were those used in preparing Table 4. The amount of window area in shadow resulting from the window setback can be calculated as suggested in THE GUIDE 1957.

The shade temperatures for a 75 F indoor temperature, after being adjusted for an outside conductance of 3 Btu per (hr) (sq ft) (F deg) in a manner similar to that

TABLE 2. . . . *K* & *U* VALUES TO BE USED IN EQUATION 1 FOR HEAT FLOW CALCULATION

SHADE MATERIAL		TYPE OF GLASS	*U*, BTU (HR) (SQ FT) (F DEG)	*K* VALUES				
GLASS SIDE	ROOM SIDE			INCIDENCE ANGLE OF DIRECT RADIATION[a]				DIFFUSE RADIATION
				0–50	60	70	80	
D. Green	D. Green	Regular Plate	0.76	0.52	0.49	0.43	0.30	0.49
D. Green	Al. Foil	"	0.70	0.48	0.46	0.40	0.28	0.45
Al. Foil[b]	Al. Foil	"	0.57	0.22	0.22	0.20	0.15	0.21
White (opaque)	White (opaque)	"	0.76	0.20	0.19	0.17	0.13	0.18
White (Transl.)	White (Transl.)	"	0.76	0.28	0.26	0.23	0.16	0.25
Al. Pigm.	White	"	0.65	0.30	0.34	0.33	0.24	0.30
D. Green	D. Green	Heat Abs.	0.76	0.39	0.38	0.33	0.28	0.38
White (opaque)	White (opaque)	"	0.76	0.24	0.23	0.21	0.18	0.23
White (Transl.)	White (Transl.)	"	0.76	0.29	0.28	0.24	0.20	0.28

[a] Angle between the normal to the surface and the sun's direct beam.
[b] With a bright foil, solar absorptance of about 0.2, *K* values are about 0.15.
Note: (1) *K* and *U* values are adjusted for an outside conductance of 3 Btu per (hr) (sq ft) (F deg) and an inside conductance as suggested in Reference 4. (2) Glass-shade spacing—3 in.

used in Appendix B, were plotted against αI_{gs}. In Fig. 4 the ordinate is the shade temperature in the presence of solar radiation adjusted to a zero temperature difference between indoor and outdoor air. This is expressed by $t_s - r(t_o - 75)$, where t_s is the shade temperature resulting from both solar radiation and outdoor-indoor air temperature difference, and r is a factor for adjusting t_s to zero outdoor-indoor temperature difference. The r is defined as the ratio of the temperature differences between the shade and indoor air temperatures, and the outdoor air and indoor air temperatures. It was determined from the tests made with no solar radiation.

For a given glass and roller-shade combination r was constant over a wide range of outdoor air temperatures. With no solar radiation, both $t_s - r(t_o - 75)$ and αI_{gs} are zero, and the curves in Fig. 4 pass through the origin. As seen in Fig. 4, the data for each glass and roller-shade combination correlated with a straight line passing through the origin; and knowing the slope of the line, r and αI_{gs}, the shade temperature can be calculated. For practical purposes, however, αI_{gs} is not an easy term to calculate. Therefore, by means of a procedure similar to that ex-

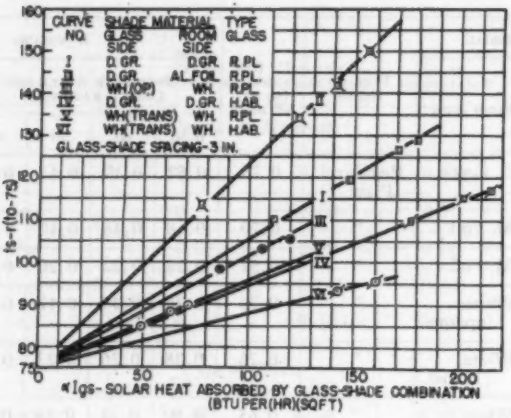
CURVE NO.
SHADE MATERIAL GLASS SIDE
TYPE ROOM GLASS SIDE
I D.GR. D.GR. R.PL.
II D.GR. AL.FOIL R.PL.
III WH.(OP) WH. R.PL.
IV D.GR. D.GR. H.AB.
V WH.(TRANS) WH. R.PL.
VI WH.(TRANS) WH. H.AB.
GLASS-SHADE SPACING-3 IN.
$t_s - r(t_o - 75)$
αI_{gs} - SOLAR HEAT ABSORBED BY GLASS-SHADE COMBINATION (BTU PER (HR)(SQ FT)

FIG. 4. . . . ROLLER-SHADE TEMPERATURE DUE TO SOLAR RADIATION

plained in Appendix C, the shade temperature for a 75 F indoor temperature can be expressed by Equation 2, thus:

$$t_s = SI_v + r(t_o - 75) + 75 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

S = a factor depending upon whether the solar radiation is diffuse or direct, the incidence angle of the direct beam with the glass, and the type of glass and roller-shade combination.

For the glass and roller-shade combinations tested, S and r values to be used in Equation 2 are given in Table 6.

APPLICATION OF DATA

The summer heat gains through a glass and roller-shade combination can be calculated from the data in Tables 4 and 5. The former table is for the heat gains through the sunlit section and the latter table for the section in shadow caused by the window setback. The amount of window area in the shadow can be determined as suggested in THE GUIDE 1957 (p. 320). It is to be noted that Tables 4 and 5 are

for an inside temperature of 75 F, and for outside temperatures as indicated in the table for different hours of the day. For temperature differentials other than those used in these tables, the data may be corrected by adding or subtracting U Btu per (hr) (sq ft) for each degree increase or decrease in temperature differential. The U values are given in Table 2.

Although the Table 4 data for the heat gains with roller-shades are for regular plate and heat absorbing glass only, the data for a regular plate glass can be used

TABLE 3. . . . U VALUES FOR A ROLLER-SHADE AND A SINGLE GLASS COMBINATION (WINTER DATA WITH NO SOLAR RADIATION)

SHADE MATERIAL		OPENING	U[a] BTU (HR) (SQ FT) (F DEG)
GLASS SIDE	ROOM SIDE		
		Fully Drawn Shade:	
Cloth	Cloth	½-In. opening all around the edges	0.76
Al. pigm.	Cloth	½-In. opening all around the edges	0.65
Cloth	Al. foil	½-In. opening all around the edges	0.70
Al. foil	Al. foil	½-In. opening all around the edges	0.57
Cloth	Cloth	All edges sealed	0.53
Al. foil	Al. foil	All edges sealed	0.22
		Half Drawn Shade:	
Cloth	Cloth	½-In. opening all around the edges	0.88
		No shade	0.95[b]

[a] U values adjusted for an outside conductance of 3 Btu per (hr) (sq ft) (F deg) and an inside conductance as suggested in Reference 4.
[b] Guide value 1.13 adjusted for outside conductance of 3 Btu per (hr) (sq ft) (F deg).
Note: Cloth shades vinyl coated.

for ordinary window glass. In such a case, heat gains will be about 5 percent higher with dark green shades and 8 percent lower with white shades than the corresponding values in Table 4 for the regular plate glass.

For outdoor conditions other than those shown in Tables 4 and 5, the heat flow through a glass and roller-shade combination can be calculated from Equation 1, and the K and U values for this equation can be taken from Table 2.

For a given outdoor condition, the shade temperature can be calculated from Equation 2, and the S and r values for this equation can be taken from Table 6.

Example: A westerly oriented, 3-ft wide × 5-ft high regular plate glass window is covered by a fully drawn dark green roller-shade spaced 3 in. from the glass. The window setback is such that at 3 p.m. 18 percent of the surface of the glass is in the shadow. Calculate the heat gains at 3 p.m. from Tables 4 and 5 for (1) 75 F indoor and (2) 80 F indoor temperatures. (3) Calculate the roller-shade temperature at 3 p.m. for 75 F indoor temperature.

Given: THE GUIDE *values for the solar radiation intensities at 3 p.m. are: Direct (normal) 267 Btu per (hr) (sq ft), diffuse on a west wall, 35 Btu per (hr) (sq ft), incidence angle on a west wall 47.7 degree, and cosine of the incidence angle is 0.673.*

Solution: The window area in shadow is $3 \times 5 \times 0.18 = 2.7$ sq ft and the sunlit area is $3 \times 5 - 2.7 = 12.3$ sq ft.

1. For 75 F indoor temperature: The heat gain for the sunlit section (from Table 4, column 4) $12.3 \times 125 = 1537.5$ Btu per hr; for the section in shadow (from Table 5, column 4) $2.7 \times 32 = 86.4$ Btu per hr. The total heat gain is $1537.5 + 86.4 = 1624$ Btu per hr.

2. For 80 F indoor temperature: Tables 4 and 5 data for 3 p.m. are based on the air temperature difference $95 - 75 = 20$ F deg, whereas for this example it is $95 - 80 = 15$ F deg. Allowing $U = 0.76$ (from Table 2) Btu per (hr) (sq ft) the correction per Fahrenheit degree difference in temperature differential, the reduction in the heat gain is $15 \times 5 \times 0.76 = 57.0$ Btu per hr. Hence, the heat gain is $1624 - 57.0 = 1567$ Btu per hr.

TABLE 4. . . .HEAT GAINS THROUGH THE SUNLIT SECTION OF GLASS AND ROLLER-SHADE COMBINATION FOR 75 F INSIDE TEMPERATURE

ORIENTATION	SUNTIME	OUTSIDE TEMP—F DEG	HEAT GAINS—BTU PER (HR) (SQ FT)								
			REGULAR PLATE GLASS WITH ROLLER-SHADES			HEAT ABSORBING GLASS WITH ROLLER-SHADES			GLASS ONLY		
			DARK GREEN	WHITE TRANSL	WHITE OPAQUE	DARK GREEN	WHITE TRANSL	WHITE OPAQUE	ORD. WINDOW	REG. PLATE	HEAT ABS.
1	2	3	4	5	6	7	8	9	10	11	12
East	6 am	74	77	40	28	57	43	35	128	118	85
	7	75	120	63	45	90	67	56	199	183	135
	8	77	126	67	49	95	71	60	211	192	142
	9	80	114	62	46	88	66	56	187	172	129
	10	83	85	48	38	67	50	44	137	124	94
	11	87	47	29	25	40	31	28	71	67	54
	12	90	23	17	16	20	18	17	33	32	27
South	8 am	77	13	7	6	10	8	7	20	19	14
	9	80	31	18	14	25	19	16	48	44	34
	10	83	51	30	24	42	32	28	78	73	57
	11	87	67	40	32	54	41	37	104	96	75
	12	90	74	44	36	59	46	41	115	107	84
	1 pm	93	71	44	37	58	46	41	110	103	82
	2	94	59	38	33	50	40	36	90	85	70
	3	95	42	29	25	36	30	28	63	59	50
	4	94	26	20	19	23	21	20	37	36	32
West	12	90	23	17	16	20	18	17	33	32	27
	1 pm	93	52	34	29	45	36	33	77	73	61
	2	94	93	57	46	75	59	53	147	136	107
	3	95	125	74	58	99	77	68	204	189	148
	4	94	139	80	62	108	84	73	228	210	164
	5	93	133	77	58	104	81	69	217	202	157
	6	91	90	53	41	70	55	48	144	136	104

Note: (1) Data for regular plate glass can be used for ordinary window glass. (2) Glass shade spacing—3 in.

3. Roller-shade temperature for 75 F indoor temperature: The roller-shade temperature can be calculated from Equation 2 by taking S and r values from Table 6. For the direct solar radiation with 47.7 degree incidence angle, $S = 0.24$; for diffuse solar radiation $S = 0.22$, and $r = 0.27$. The intensity of the direct solar radiation on the wall is $267 \times 0.673 = 180$ Btu per (hr) (sq ft); the diffuse solar radiation is 35 Btu per (hr) (sq ft); and the outside air temperature is 95 F. Substituting these values in Equation 2, the roller-shade temperature for the sunlit section is $t_s = 0.24 \times 180 + 0.22 \times 35 + 0.27\,(95 - 75) + 75 = 131.3$ F, and for the section in the shadow, $t_s = 0.22 \times 35 + 0.27 \times (95 - 75) + 75 = 88.1$ F.

DISCUSSION OF RESULTS

The solar calorimeter investigations were made under winter and summer conditions. The winter testing, with no solar radiation at night, supplied information regarding the effect of glass to shade spacing, shade emissivity, and shade opening on the heat loss resulting from the temperature difference between the indoor and outdoor air. As no solar radiation was present during these tests, the solar absorptance

TABLE 5. . . .HEAT GAINS THROUGH GLASS AND ROLLER-SHADE COMBINATION REMAINING IN SHADOW CAUSED BY WINDOW SETBACK

(For 75 F Inside Temperature)

ORIENTATION	SUNTIME	OUTSIDE TEMP-F	HEAT GAINS BTU PER (HR) (SQ FT)		
			DARK GREEN	WHITE TRANSL	WHITE OPAQUE
1	2	3	4	5	6
East	6 am	74	10	5	3
	7	75	17	9	6
	8	77	19	11	8
	9	80	21	13	10
	10	83	21	14	12
	11	87	21	16	14
	12	90	22	17	16
South	8 am	77	11	6	5
	9	80	15	10	8
	10	83	19	13	11
	11	87	23	16	14
	12	90	26	19	17
	1 pm	93	27	21	19
	2	94	27	21	19
	3	95	26	21	19
	4	94	24	19	18
West	12	90	23	17	16
	1 pm	93	26	20	18
	2	94	29	21	20
	3	95	32	24	22
	4	94	32	23	21
	5	93	30	22	20
	6	91	23	18	16

Note: For ordinary window, regular plate, or heat absorbing glass.

of the shade material and glass were not important. With a fully drawn roller-shade having about ½-in. opening on the edges, the variation of glass to shade spacing from ¾ in. to 5¾ in. had no significant effect on the heat loss. The emissivity of the roller-shade material for long-wave radiation and amount of shade opening influenced the heat loss as shown by the U values in Table 3. The U values

TABLE 6. . . . S AND r VALUES TO BE USED IN EQUATION 2 FOR ROLLER-SHADE TEMPERATURE CALCULATIONS

SHADE MATERIAL		TYPE OF GLASS	S VALUES					
GLASS SIDE	ROOM SIDE		r	INCIDENCE ANGLE OF DIRECT RADIATION				DIFFUSE RADIATION
				0–50	60	70	80	
D. Green	D. Green	Regular Plate	0.27	0.24	0.22	0.20	0.14	0.22
D. Green	Al. Foil	"	0.31	0.38	0.36	0.31	0.22	0.35
Al. Foil[a]	Al. Foil	"	0.22	0.35	0.33	0.29	0.20	0.33
White (Opaque)	White (Opaque)	"	0.27	0.11	0.10	0.09	0.07	0.10
White (Transl.)	White (Transl.)	"	0.27	0.06	0.06	0.06	0.05	0.06
Al. Pigm.	White	"	0.24	0.15	0.17	0.17	0.12	0.15
D. Green	D. Green	Heat Abs.	0.27	0.17	0.16	0.14	0.12	0.16
White (Opaque)	White (Opaque)	"	0.27	0.11	0.10	0.09	0.08	0.10
White (Transl.)	White (Transl.)	"	0.27	0.09	0.09	0.08	0.07	0.08

[a] With a bright foil, solar absorptance of about 0.2, S values are about 0.25.
Note: S & r values are adjusted for an outside conductance of 3 Btu per (hr) (sq ft) (F deg) and an inside conductance as suggested in Reference 4.

are lower for the fully drawn aluminum foil roller-shade (*i.e.*, low emissivity) than for the cloth roller-shades. By sealing the edges of the roller-shade to the window frame the circulation of room air between the roller-shade and the glass is stopped, and therefore, the heat loss is lower than with a ½-in. space between the edges of the roller-shade and the window frame. A window having a half-drawn roller-shade will have a U value which is the average of the U values for a fully drawn shade and a window with no roller-shade.

The summer heat gains for a glass-shade combination were investigated in the presence of the solar radiation with fully drawn roller-shades having about ½-in. opening on the edges. The variation in glass-shade spacing from ¾ in. to 3 in. had

no effect on the heat gain; but with 5¾-in. spacing the heat gain was about 5 to 10 percent higher. The data presented in this paper are for a 3-in. glass-to-shade spacing.

The solar heat gain was less with a roller-shade having a low solar absorbing surface (*i.e.*, white) on the glass side. However, the heat gain was higher with a white translucent roller-shade than with a white opaque shade, although the solar absorptance of the former was less. The reason for this was that the solar energy transmitted through the translucent material offset the decrease in the convection-radiation component of the heat gain resulting from the lower solar absorption.

Lowering the emissivity of the surface of the roller-shade for long-wave radiation (*i.e.*, aluminum) on the room side reduced the heat gain due to solar radiation. However, lowering the emissivity of the surface of the shade on the glass side increased the heat gain due to solar radiation with a regular plate glass, but decreased it with a heat absorbing glass.

TABLE 7. . . .REDUCTION IN HEAT GAIN BY FULLY DRAWN ROLLER-SHADES

TYPE OF GLASS	HEAT REDUCTION BY ROLLER-SHADES, PERCENTAGE[a]					
	REGULAR PLATE			HEAT ABSORBING		
TYPE OF ROLLER SHADE	DARK GREEN	WHITE TRANSL.	WHITE OPAQUE	DARK GREEN	WHITE TRANSL.	WHITE OPAQUE
South[b]	30	56	64	28	43	49
West[c]	33	60	68	32	47	53

[a] Percentage of heat gain through unshaded glass.
[b] From 8 a.m. to 4 p.m.
[c] From 12 noon to 6 p.m.
Note: Data are for a window with no setback and based on Table 4 values.

A dark green shade in combination with heat absorbing glass gave a lower heat gain than the same roller-shade when used with regular plate glass. This was due primarily to the high heat absorption of the heat absorbing glass in excluding the initial entry of the solar heat, and to the low reflectivity of the green roller-shade. However, with a white roller-shade, heat gains were less with a regular plate glass than with heat absorbing glass. This can be explained by the fact that with the regular plate glass, the white roller-shade was able to reflect an appreciable percentage of the solar energy back to the outside. With the heat absorbing glass, although less heat was transmitted directly to the roller-shade, a sizeable portion of that which was reflected was absorbed by the glass, thus increasing the glass temperature.

It is interesting to note that the K values given in Table 2 are a relative measure of the solar heat gain with different type of glass and roller-shade combinations. The smaller the K value, the smaller is the solar heat gain.

Table 7 illustrates the effectiveness of the roller-shades in excluding solar heat over that period of the day during which the direct sun falls upon the window. It is to be noted that the data in this table are based on Table 4 values, which were for a window with no setback. For a window with a setback, however, heat exclusion is less than given in Table 7.

The long-wave radiation emitted by the room side surface of the roller-shade affects the comfort of the occupants, and the amount of radiation emitted depends on the roller-shade temperature and emissivity. The roller-shade temperature is high with a dark-colored shade and low with a white one. With an aluminum roller-shade or a dark green shade with aluminum surface on the room side, the roller-shade temperature is also high; but, with such shades long-wave radiation emission will be less because ot the low emissivity of the aluminum surface.

The convection-radiation component of heat gain for an outside conductance of 3 Btu per (hr) (sq ft) (F deg) can be calculated for any shade other than those tested by the method described in Appendix B, by using Equation B-7 and by taking the U and U' values in this equation from the curves in Fig. B-1. The test results are considered correct within 10 percent.

CONCLUSIONS

1. The summer heat gains through a glass and roller-shade combination can be obtained from the data presented in Tables 4 and 5, for east, south and west orientations and for various hours of the day. For outdoor conditions other than those given in Tables 4 and 5, the summer heat gains and winter heat losses can be calculated from Equation 1 and Table 2.

2. The shade temperatures can be calculated from Equation 2 and Table 6.

3. With an opaque roller-shade, solar heat gains are reduced by lowering the solar absorptance of the surface of the shade on the glass side.

4. Effect of a low emissivity surface for long-wave radiation on the room side of the roller-shade is to reduce the heat gain due to solar radiation.

5. Effect of a low emissivity surface on the glass side of the roller-shade is to increase the solar heat gains with a regular plate glass and decrease it with a heat absorbing glass.

6. With an opaque roller-shade having a white surface on the glass side the solar heat gains are less with regular plate glass than with heat absorbing glass. However, with a dark green roller-shade, heat gains are less with heat absorbing than with regular plate glass.

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[6] G. V. Parmelee, W. W. Aubele, and D. J. Vild: ASHVE RESEARCH REPORT No. 1474—The shading of sunlit glass—An experimental study of slat-type sun shades (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 221).

APPENDIX A

CONVECTION-RADIATION HEAT GAIN

Consider a glass and roller-shade combination with no solar radiation. The heat flow through the window is

$$\text{Heat flow} = U(t_i - t_o) = h_o(t'_g - t_o) \quad \text{(A-1)}$$

where,

t'_g = the glass temperature with no solar radiation.

With the solar radiation, however, the solar energy absorbed by the glass and roller-shade increases their temperatures until a balance is reached between the rate of heat absorption by them and the rate of heat dissipation from them by convection and radiation. Assuming a negligible storage, the following energy balance equations can be written for the glass and shade:

$$\alpha I_g = h_o(t_g - t_o) + C_{gs} + R_{gs} \quad \text{(A-2)}$$

$$\alpha I_s = CR_{si} + C_{ss} + R_{sg} \quad \text{(A-3)}$$

where,

t_g = glass temperature with the solar radiation.

Adding these two equations

$$\alpha I_g + \alpha I_s = h_o(t_g - t_o) + (CR_{si} + C_{gs} + C_{ss}) \quad \text{(A-4)}$$

In Equation A-4, $(\alpha I_g + \alpha I_s)$ is the solar energy absorbed by the glass and shade, and $(CR_{si} + C_{gs} + C_{ss})$ is the total convection-radiation heat gain into the room. Let,

$$\alpha I_g + \alpha I_s = \alpha I_{gs} \text{ and } CR_{si} + C_{gs} + C_{ss} = CR \quad \text{(A-5)}$$

Substituting these in equation A-4,

$$\alpha I_{gs} = CR + h_o(t_g - t_o) \quad \text{(A-6)}$$

Substituting $t_g = t'_g + \Delta t$ in Equation A-6

$$\alpha I_{gs} = CR + h_o(t'_g - t_o) + h_o\Delta t \quad \text{(A-7)}$$

From Equations A-1 and A-7,

$$CR + U(t_i - t_o) = \alpha I_{gs} - h_o\Delta t \quad \text{(A-8)}$$

In Equation A-8, Δt is the rise in glass temperature due to the solar radiation; and for a given indoor condition and outside conductance it is a function of αI_{gs}. This suggests

that, for a given indoor condition and outside conductance $CR + U(t_i - t_o)$ should be a function of αI_{ss}.

APPENDIX B

EFFECT OF OUTDOOR CONDUCTANCE AND SHADE EMISSIVITY ON THE SOLAR HEAT GAIN

Consider a combination of a glass and a roller-shade with sealed edges in the presence of the solar radiation. With negligible storage, the energy balance equations for the glass and shade with sealed edges, (*i.e.* no air circulating from the glass-to-shade spacing into the room) are:

$$\alpha I_g = h_o(t_g - t_o) + h_a(t_g - t_s) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(B-1)}$$

$$\alpha I_s = CR + h_a(t_s - t_g) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(B-2)}$$

From Equations B-1 and B-2, eliminating t_g,

$$\left(\frac{1}{h_o} + \frac{1}{h_a}\right) CR + t_s = \frac{1}{h_o}\alpha I_g + \left(\frac{1}{h_o} + \frac{1}{h_a}\right)\alpha I_s + t_o \quad . \quad . \quad . \quad \text{(B-3)}$$

where, CR and t_s = interdependent variables.

For the solar calorimeter, the relation between CR and t_s was given in Reference 5. For the design condition, however, this relation, as suggested in Reference 4, was:

$$CR = 0.27(t_s - t_i)^{1.25} + \xi(E_s - E_i) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(B-4)}$$

For given values of t_i, h_o and h_a, it is possible to plot CR as a function of the composite term on the right hand side of Equation B-3, (*i.e.* $1/h_o\,\alpha I_g + (1/h_o + 1/h_a)\alpha I_s + t_o$) from the foregoing relations. This was plotted for 75 F indoor temperature and a 3-in. air space between glass and shade, for the different values of h_o, by taking the air space conductances, h_a, the same as those given in THE GUIDE 1958, p. 176. The curves obtained in this manner were almost straight lines. Hence, the straight line equation for CR with a sealed roller-shade is:

$$CR = U'\left[\frac{1}{h_o}\alpha I_g + \left(\frac{1}{h_o} + \frac{1}{h_a}\right)\alpha I_s + t_o - 75\right] \quad . \quad . \quad . \quad . \quad \text{(B-5)}$$

and for a 75 F outdoor temperature,

$$CR = U'\left[\frac{1}{h_o}\alpha I_g + \left(\frac{1}{h_o} + \frac{1}{h_a}\right)\alpha I_s\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(B-6)}$$

where,

U' = the solar heat transfer coefficient.

The convection-radiation heat gains from the test data for the shade with unsealed edges, after being adjusted for a 75 F outdoor temperature, were compared with those obtained from Equation B-6. For a roller-shade with a given emissivity, the ratio of CR from the test data and from Equation B-6 was found to be constant within 10 to 15 percent, regardless of the solar absorptance of the shade material. Therefore, a variation in h_o alters the solar heat gain for the test data in the same proportion as it alters CR defined by Equation B-6. This relation was used for adjusting the solar heat gains for the desired outside conductance.

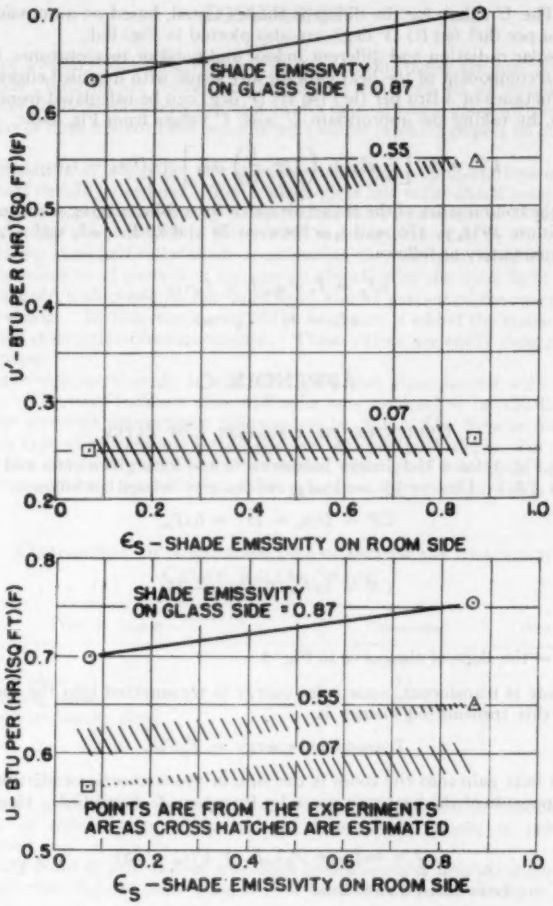

FIG. B-1. . . . U AND U' VALUES TO BE USED IN EQUATION B-7, THAT IS, $CR = U' [(\alpha I_g/3) + (1/3 + 1/h_a)\alpha I_s] + U(t_o + t_i)$ WHERE t_i IS BETWEEN 70 AND 80 F

The values of U for the shades tested with unsealed edges, as obtained from Equation B-6, are given in Fig. B-1 as a function of the shade emissivity. The U values given in this figure, are for an outside conductance of 3 Btu per (hr) (sq ft) (F deg) and for equal outdoor and indoor air temperatures. The extended range of values, although only approximate, serve to indicate the effect of shade emissivity on the solar heat gain.

With no solar radiation, the heat flow is due to the indoor and outdoor air temperature difference, and the overall coefficient of heat transfer (*i.e.* U) is the factor controlling the

heat flow. The U values for the different shades tested, based on an outside conductance of 3 Btu per (hr) (sq ft) (F deg), are also plotted in Fig. B-1.

With the solar radiation and different indoor and outdoor temperatures, the convection-radiation component of the heat gain for any shade with unsealed edges and for an outside conductance of 3 Btu per (hr) (sq ft) (F deg) can be calculated from the following equation, by taking the appropriate U' and U values from Fig. B-1:

$$CR = U'\left[\frac{1}{3}\alpha I_g + \left(\frac{1}{3} + \frac{1}{h_a}\right)\alpha I_a\right] + U(t_o - t_i) \quad \text{(B-7)}$$

where, h_a is the conductance of the sealed air space for a 3-in. spacing, and may be taken from THE GUIDE 1958, p. 176, and t_i is between 70 and 80 F. αI_g and αI_a can be calculated approximately as follows:

$$\alpha I_g = I_v \cdot \alpha_g(1 + T_g\gamma_a) \quad \text{(B-8)}$$

$$\alpha I_a = I_v \cdot T_g \cdot \alpha_a \quad \text{(B-9)}$$

APPENDIX C

DERIVATION OF EQUATION 1 OF PAPER

As seen in Fig. 3, for a 75 F indoor temperature and for a given glass and roller-shade combination $CR - U(t_o - 75)$ and αI_g, are linearly related as follows:

$$CR - U(t_o - 75) = b\alpha I_g \quad \text{(C-1)}$$

or

$$CR = b\alpha I_g + U(t_o - 75) \quad \text{(C-2)}$$

where

b = the slope of the curve in Fig. 3.

If the shade is translucent, some solar energy is transmitted into the room, and the amount of this transmitted energy is,

$$\text{Transmitted energy} = T_g T_s I_v \quad \text{(C-3)}$$

The total heat gain into the room is the sum of the convection-radiation and transmitted components of the heat gain given by Equations C-2 and C-3. Hence, the total heat gain is:

$$Q = b\alpha I_g + T_g T_s I_v + U(t_o - 75) \quad \text{(C-4)}$$

where, αI_g can be evaluated as follows

$$\alpha I_g = (\alpha_g + T_g\alpha_a + T_g\gamma_a\alpha_g)I_v \quad \text{(C-5)}$$

substituting C-5 in C-4,

$$Q = [b(\alpha_g + T_g\alpha_a + T_g\gamma_a\alpha_g) + T_g T_s]I_v + U(t_o - 75) \quad \text{(C-6)}$$

$$\text{Let } K = b(\alpha_g + T_g\alpha_a + T_g\gamma_a\alpha_g) + T_g T_s \quad \text{(C-7)}$$

$$\text{Then } Q = K_y I_v + U(t_o - 75) \quad \text{(C-8)}$$

where, the K-factor depends on whether the solar radiation is diffuse or direct, the incidence angle of the direct beam, the solar properties of the glass and shade, and an experimentally determined value, b.

It will be seen that Equation C-8 is the same as Equation 1 of the text.

DISCUSSION

D. J. VILD, Toledo, Ohio (WRITTEN): The data in this paper are well presented and easily applied. Information contained therein may be extended to include insulating glass and roller-shade combinations as well as any type of single glass with roller-shades. This extension of data has not been possible with earlier research papers on other shading devices.

Using information in this paper, the discusser made some computations using fundamental thermal circuit procedures. For a single-glass and roller-shade combination the heat gain due to absorbed solar energy is determined to be equivalent to 25 percent of the energy absorbed by the glass and 73 percent of the energy absorbed by the roller-shade. For an insulating glass and roller-shade combination the heat gain from absorbed solar energy is equivalent to 15 percent of the energy absorbed by the outer light of glass, 57 percent of the energy absorbed by the inner light, and 85 percent of the energy absorbed by the roller-shade. To this component of the heat gain is added the transmitted solar energy and the convection-conduction gain. These values are easily determined using data in this paper.

Some comparisons were made of unshaded glass and glass shaded with dark green roller-shades. The total incident solar radiation was assumed to be 150 Btu per (hr) (sq ft) and the air-to-air temperature difference to be 20 F. The *U*-value for insulating glass with this type of roller-shade is 0.46 Btu per (hr) (sq ft) (F deg). For purposes of easy comparison the heat gain through unshaded plate glass was made equal to 100 and other values changed accordingly. The values are tabulated in Table A.

TABLE A. . . .COMPARISON OF UNSHADED GLASS AND GLASS SHADED WITH DARK GREEN ROLLER-SHADE

TYPE OF GLASS	UNSHADED	WITH SHADE
Regular plate	100	67
Heat-absorbing plate	75	54
Regular insulating glass	82	64
Heat-absorbing insulating glass	58	46

As the authors point out in the paper these comparisons cannot be expected to hold for all types of roller-shades. The type of roller-shade chosen is probably the most widely used.

An interesting point to note is that unshaded heat-absorbing insulating glass shows a lower heat gain than regular plate glass with a fully drawn roller-shade.

G. R. MUNGER, Toledo, Ohio: It is urged that there be early correlation of the *K*-factor developed in recent work at the Laboratory and the shade factors in THE GUIDE so that people in the field can relate shade factors to the *K*-factor.

R. W. MCKINLEY, Pittsburgh, Pa.: The authors assisted the casual reader by carrying out certain calculations which illustrate values of a correction factor that might give an idea of how these are related. It is suggested that the authors also provide a similar illustration of the temperature of the shade. This might help the casual reader greatly.

AUTHORS' CLOSURE (Mr. Ozisik): The authors agree with the results of comparison made by Mr. Vild regarding heat gains through shaded and unshaded glass with dark green roller-shade as shown in Table A. However, the results will vary with the type of shade used in combination. For instance, with a white opaque roller-shade the figures

under the column "With Shade" would be from 40 to 32 percent, as compared with Mr. Vild's results from 67 to 46 percent with a dark green roller-shade.

Mr. McKinley suggested inclusion of illustration of the temperature of the shade. For instance, using the same outdoor and indoor conditions as the Table 4 heat gain values were based on, the shade temperature with regular plate glass for west orientation at 4 p.m. would be 106 F with white opaque, 137 F with dark green and 162 F with aluminum shades. It is, however, to be noted that the amount of radiation emitted by the room side surface of the roller-shade into the room would depend upon not only the temperature of the shade, but also upon its emissivity for long-wave radiation on the room side.

Mr. Munger suggests relating the *K*-factors to the shade factors now in THE GUIDE. Actually, attention is being directed toward compiling all the data on shading devices and expressing them on a comparable basis.

1697

No. 1697

EVALUATION OF THREE ROOM AIR DISTRIBUTION SYSTEMS FOR SUMMER COOLING

By J. J. Reinmann*, Alfred Koestel**, and G. L. Tuve†, Cleveland, Ohio

This paper is the result of research sponsored by the American Society of Heating, Refrigerating and Air-Conditioning Engineers, in cooperation with Case Institute of Technology, Cleveland, Ohio

THIS PAPER describes the performance of 3 residential air-distribution systems during summer cooling and isothermal operations. The object was to evaluate the room air conditions in terms of human comfort and to study in detail the air-flow phenomena. This is a continuation of an investigation reported earlier[1] in which the same 3 systems were used to study air distribution for winter heating, and in which special attention was paid to the winter operations requiring high air volumes at lower supply temperatures, as in heat pump systems. Performance under summer and winter conditions are compared in the present paper.

Comfort Criteria

The term *draft* is here defined as any localized feeling of coolness or warmth of any portion of the body, due to both air movement and air temperature, with humidity and radiation considered constant. The warmth or coolness of a draft was measured above or below the room control condition, and the latter was maintained at 76 F dry-bulb, air movement approximately 30 fpm and located at the center of the room, 30 in. above the floor.

An approximate equation was used to define the difference in effective temperature between any point, x, in the occupied zone of the test room and the control condition, *viz.*:

$$\Delta t = (t_x - 76) - 0.07(V_x - 30) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

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** Associate Professor of Heat-Power Engineering, Case Institute of Technology. Member of ASHRAE.
† Professor of Mechanical Engineering, Case Institute of Technology. Member of ASHRAE.
[1] Exponent numerals refer to References.
Presented at the Annual Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Lake Placid, N. Y., June 1959.

The form of this equation was originally proposed by Rydberg and Norback[2]. Equation 1 differs from that used in the earlier paper of this series by the addition of 30 fpm at the control point. This change was recommended by Straub, in his discussion[3]. Equation 1 takes into account the feeling of coolness produced by air motion. In summer the local air-stream temperature t_x is below the control temperature, hence both temperature and velocity terms are negative; *i.e.*, both of them add to the feeling of coolness. If, in winter, t_x is above the control temperature, any appreciable air velocity will subtract from the feeling of warmth produced by

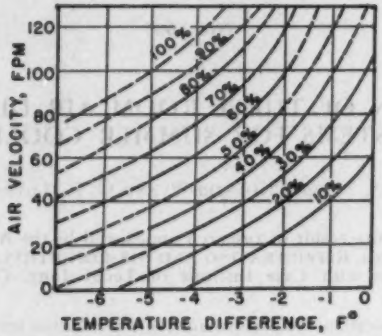

FIG. 1. . . . PERCENTAGE OF NORMAL OCCUPANTS OBJECTING TO DRAFTS IN AIR-CONDITIONED ROOMS (NECK REGION)

t_x. Thus it is usually possible to have zero difference in effective temperature between location x and the control point in winter, but not in summer.

Houghten[4] presented data which make it possible to interpret statistically the percentage of room occupants that will tolerate or object to a given draft condition. Fig. 1 presents Houghten's data, with some extrapolation as required for these studies.

The *occupied zone* was arbitrarily defined as the region in the test room enclosed by imaginary planes 1 ft from each of the 4 walls and 6 ft 2 in. from the floor. Since the air must enter the occupied zone through these imaginary boundaries, the most severe conditions of high air velocity and low temperature were assumed to be found somewhere along these boundaries. Therefore all measurements to locate the *critical draft regions* were made at the boundaries of the occupied zone.

Three comfort criteria were used to evaluate the room air distribution:

1. Draft, *i.e.*, the difference between the local air temperature and velocity and those at the control point.

2. Excessive local air velocities causing movement of hair, eye discomfort, disturbance of paper, fabrics, etc.

3. Vertical air temperature gradients in the occupied zone, causing discomfort due to differences in warmth between head and ankles.

With respect to these 3 criteria, the conditions of Table 1 were selected as the maximum allowable departures within the occupied zone, from the comfort or ideal situation, for summer conditions.

GENERAL CONCLUSIONS

Under the rigid comfort requirements just described, *i.e.*, at least 90 percent of the occupants comfortable in *any* part of the occupied zone, there were marked differences in the results with these 3 systems of air distribution. The air-flow patterns, air supply volume and cooling load were measured and correlated with comfort conditions, for a variety of simulated outdoor temperatures up to 100 F. The high sidewall system would meet the requirements of Table 1 only if the cooling loads and air-supply volumes were low. The system of baseboard radial diffusers, while it produced good mixing of supply and room air with high rates of air supply, exceeded the specified limits of Table 1 from the standpoint of drafts. Excessive air motion was caused in some parts of the room, while there was stagnant air in other parts. The system using ceiling diffusers was capable of satisfactory operation over a wide range of loads. It was therefore more satisfactory than either of the others from the comfort standpoint.

TABLE 1—ARBITRARY COMFORT LIMITS

1. Draft, a maximum of 10 percent of the subjects objecting, *i.e.*, 90 percent tolerability, Fig. 1.
2. Air velocity, a maximum of 50 fpm.
3. Vertical air temperature difference from floor to ceiling of occupied zone, 3 deg maximum.

With the high sidewall system, the general pattern of air motion was almost independent of the air-flow rate. When the system with baseboard radial diffusers was operated with an air-flow rate below a critical value corresponding to about 11 air changes per hr, the relatively large gravity forces spilled the cool air into the occupied zone and produced objectionable conditions, and even at higher flow rates the local drafts would be objectionable to about 30 percent of the occupants. With the ceiling diffusers there were objectionable drafts at high flow rates, but since these units make good use of the unoccupied zone near the ceiling for air mixing and entrainment, satisfactory operation was possible at lower flow rates and correspondingly lower air-supply temperatures. With the ceiling diffusers the air-flow pattern was again almost independent of air-flow rate, since gravity forces played a minor role.

In general, all 3 systems responded to the principle that local air velocities in the room are straight-line functions of the air supply volume or number of air changes. Excessive volumes, with smaller temperature difference, were likely to produce too high an air movement throughout the conditioned space; *i.e.*, most of the room became drafty. On the other hand, the absorbing of a given cooling load by supplying small volumes of low temperature air tended to produce *spot* cooling, and discomfort by the gravity drop of slow-moving air jets. There was an intermediate *best rate*

of air flow or air change for each system. With low air change, the draft objectionability increased due to low temperature, at high air change it increased due to high velocities. But even the *best* conditions were not always satisfactory. If any best rate of air circulation could be specified for such a variety of conditions it would be around 10 or 12 air changes per hour.

TEST ROOM AND INSTRUMENTS

The test room was 18 x 13 x 8 ft, built to simulate a well-insulated living room, unfurnished, with 2 walls exposed to summer weather conditions. The exposed walls contained 3 large windows. The attic was insulated but not cooled while the

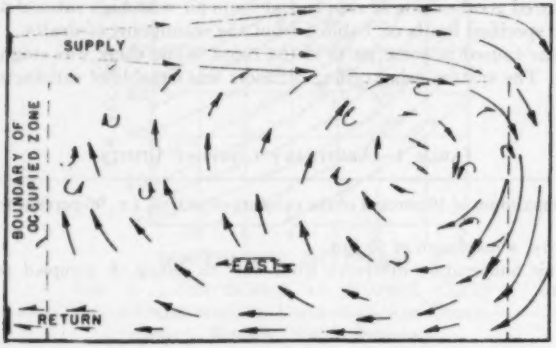

FIG. 2. . . .AIR-FLOW PATTERNS IN TEST ROOM; HIGH SIDEWALL GRILLES, ISOTHERMAL VENTILATION CONDITIONS

insulated floor had a ventilated crawl space. The room was built in a large laboratory, and since the unexposed walls were subjected to summer temperatures, they were covered with 4 in. of insulation. The heat gain from these insulated walls was around 10 percent of the total heat load. This problem did not exist in the earlier heating tests, but otherwise the room was the same as previously described[1].

Two air-velocity meters were used: (1) a modified commercial instrument of the heated-thermocouple type, and (2) a modified form of the anemometer developed by Nottage at the ASHAE Laboratory[5]. General air movement was indicated with the aid of a smoke gun and by the angular deflections of calibrated air-movement threads, located in square-grid arrangement on 6-in. centers over the boundaries of the occupied zone, as described in the earlier paper[1]. All anemometers were calibrated in a bell-mouth intake of 12-in. size or larger. Floor-to-ceiling temperature gradients were obtained from thermometers and thermocouples on vertical poles. Temperature measurements on wall, window, floor and ceiling surfaces were obtained from fine-wire thermocouples, with the reading stations outside the room. Isothermal operation was included in the tests, to check the conditions that prevail in a residential air-distribution system when cooling or heating is turned off but the blower continues to operate. All tests were run under

equilibrium conditions. At a given outdoor temperature the air-flow rate was varied to carry the load at each of several air-supply temperatures. The control condition, in the center of the room at the 30-in. level, was always maintained at 76 F.

TEST RESULTS

Results of the extensive tests are presented separately for each of the 3 systems, in the following sections. Since previous tests had shown that the location of the

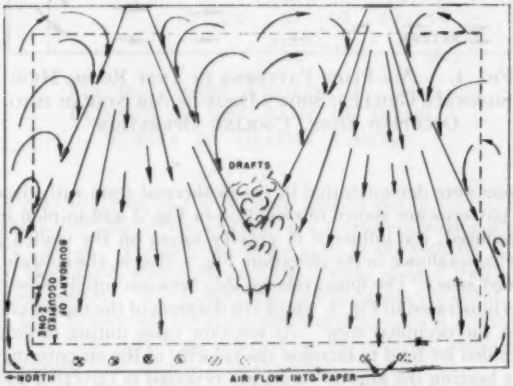

FIG. 3. . . .AIR-FLOW PATTERNS IN TEST ROOM; HORIZONTAL SECTION THROUGH TWO 12- X 6-IN. HIGH SIDEWALL GRILLES

returns had little effect on the quantities being measured[1], the only tests reported here were made with a single 18 x 8-in. baseboard return located on one of the inside walls, but the effect of return location was checked. Each set of results is presented first in terms of the general air flow pattern produced by the distribution system, the location of critical draft regions, and the relation of supply temperature and quantity to cooling load when the outdoor temperature remains constant. The system is then discussed from the standpoint of the 3 comfort criteria, *i.e.*, drafts, excessive air velocities, and objectionable floor-to-ceiling temperature gradients.

SYSTEM NO. 1, HIGH SIDEWALL SUPPLY OUTLETS

Flow Pattern, Drafts and Cooling Load: In evaluating flow patterns it should be noted that as long as the inertia forces of forced convection predominate over the gravity forces produced by temperature difference, the circulation velocities in the occupied zone will be directly proportional to the volume flow through the supply openings. The general air flow pattern will depend on the system geometry, (room space and supply outlets), and will be practically independent of the air-flow rate.

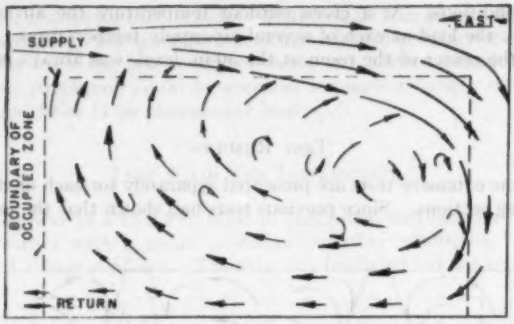

FIG. 4. . . .AIR-FLOW PATTERNS IN TEST ROOM; HIGH SIDEWALL GRILLES; SHOWS DROP OF AIR STREAM INTO OCCUPIED ZONE, COOLING OPERATION

These conditions were demonstrated in the isothermal tests with these outlets, for which the air patterns are shown in elevation in Fig. 2 and in plan in Fig. 3. As cooling is introduced, the influence of gravity forces on the chilled jet alters the pattern somewhat as shown in the elevation, Fig. 4. that is, the cold air tends to *drop* into the occupied zone. The linear relationship between supply-flow rate and room air velocities is illustrated in Fig. 5, which is a diagram of the maximum air velocities to be found in the occupied zone. At low-flow rates during cooling the gravity forces in the chilled jet tend to increase the velocity of the air entering the occupied zone. During heating the gravity forces are reversed in direction and the velocity of air entering the occupied zone is therefore decreased. As expected, the velocities during isothermal operation are intermediate between those for heating and those for cooling.

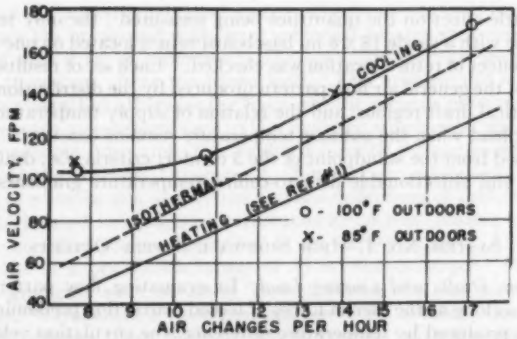

FIG. 5. . . .MAXIMUM DRAFT VELOCITIES WITH HIGH SIDEWALL GRILLES. CEILING OF OCCUPIED ZONE (FIG. 4)

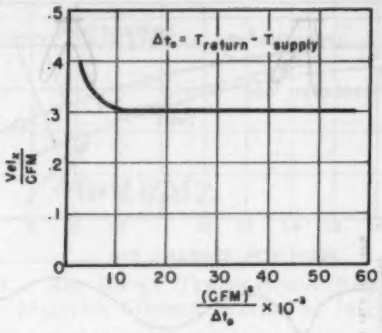

FIG. 6. . . .ROOM VELOCITY V_x AS A FUNCTION OF THE RATIO OF INERTIA FORCES TO GRAVITY FORCES

The effect of gravity forces on the room air motion can be further illustrated by Fig. 6. The ratio $cfm^2/\Delta t_o$ on the abscissas is a factor proportional to the ratio of inertia to gravity forces. The ordinate is the ratio of the velocity V_x at any point in the room to the supply flow rate, and this should remain constant as long as inertia forces predominate. For cooling, the increased gravity effects at low values of the abscissa tend to increase the room draft velocities for a given outdoor load, as shown by Fig. 6. The maximum velocities in the occupied zone occurred at the ceiling of the occupied zone, as would be expected.

The foregoing results were all obtained with two 12- x 6-in. grilles, the vertical deflecting vanes in the grilles being set to produce a uniform spread of about 60 deg

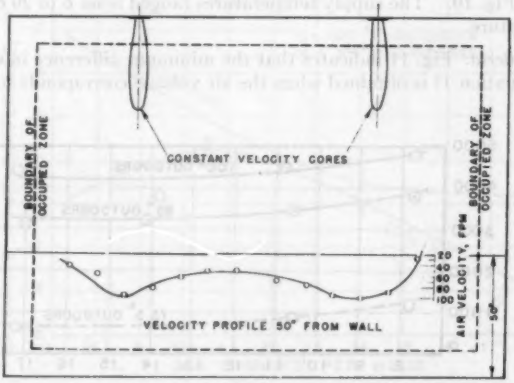

FIG. 7. . . .SIDEWALL GRILLES AS USED; 60 DEG VANE ANGLE. ISOVELS AT GRILLE LEVEL AND VELOCITY PROFILE AT 74 IN. FROM FLOOR; 8 AIR CHANGES PER HR

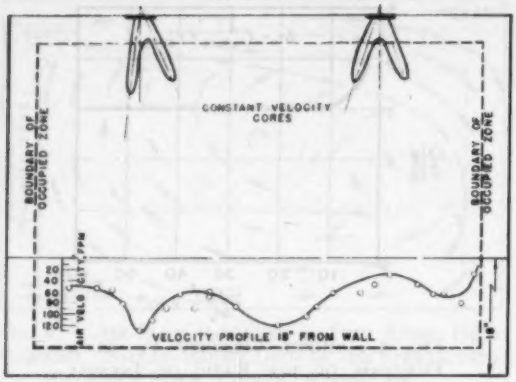
CONSTANT VELOCITY CORES
BOUNDARY OF OCCUPIED ZONE
BOUNDARY OF OCCUPIED ZONE
AIR VELO VELOCITY, FPM
VELOCITY PROFILE 18" FROM WALL

FIG. 8. . . .VELOCITY PROFILE AND ISOVELS FOR HIGH SIDEWALL GRILLES WITH WIDE-ANGLE SETTING OF VERTICAL VANES

total angle. A typical velocity profile measured at the top of the occupied zone (6-ft 2-in. level) is shown in plan in Fig. 7. When the vanes were set for a considerably wider spread at the grille, the typical bifurcated patterns were produced[6], and the corresponding velocity profile is shown in Fig. 8.

The cooling load required to maintain the control station at 76 F is shown in Fig. 9, for 3 outdoor temperatures. It is noted that there is a small but definite increase in the heat transfer through exposed surfaces at the lower air supply temperatures, in spite of the lower velocities accompanying the reduced volume required to maintain the control condition. The corresponding air-supply temperatures are shown in Fig. 10. The supply temperatures ranged from 6 to 20 deg below the room temperature.

Comfort Criteria: Fig. 11 indicates that the minimum difference in effective temperature (Equation 1) is obtained when the air volume corresponds to about 11 air

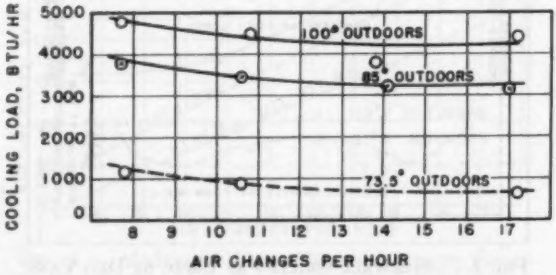

FIG. 9. . . .COOLING LOAD TO MAINTAIN CONTROL STATION AT 76 F; HIGH SIDEWALL GRILLES

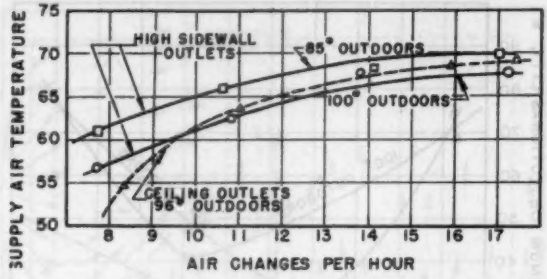

FIG. 10. . . .AIR SUPPLY TEMPERATURES REQUIRED TO MAINTAIN CONTROL STATION AT 76 F

changes per hr. At higher or lower rates of air supply this effective-temperature difference in the critical draft region increases. It would be expected from Figs. 5, 7 and 11 that a rather large percentage of occupants would object to the drafts at the 6-ft level with high sidewall outlets. Fig. 12 shows the percentage objecting, as evaluated from Fig. 1. With isothermal operation and 8 to 10 air changes per hr, fewer than 10 percent of the occupants will object to these drafts, but when operating with summer cooling the objectionability will exceed 30 percent, even at the most advantageous rate of air supply.

Vertical temperature gradients in the occupied zone were small, and did not exceed the 3 deg prescribed in Table 1 except at low air changes. In the previous winter heating tests on the same sidewall system, gradients of 10 deg or even more were observed.

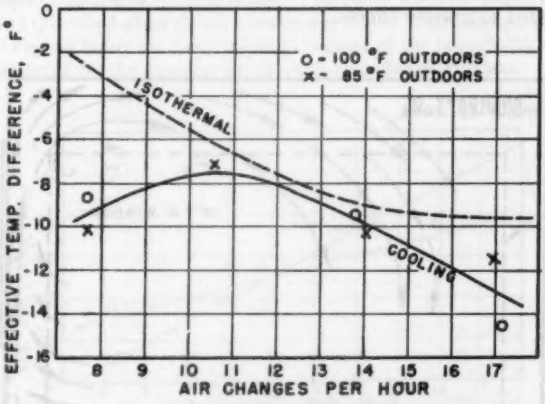

FIG. 11. . . .DIFFERENCE IN EFFECTIVE TEMPERATURE BETWEEN CONTROL STATION AND CRITICAL DRAFT REGION; SIDEWALL GRILLES

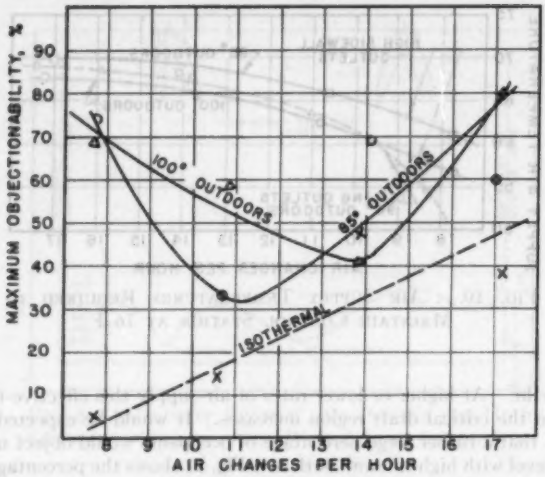

FIG. 12. . . .OBJECTIONABILITY OF DRAFTS AT VARIOUS COOLING LOADS; HIGH SIDEWALL GRILLES; CEILING OF OCCUPIED ZONE

Conclusions for Sidewall Grilles: The general pattern of room air motion remained essentially independent of air-flow rate. The critical draft region occurred at the 6-ft 2-in. level, 9 to 11 ft from the outlets. At any given air-supply flow rate, the air velocities in the critical draft region were highest during cooling, lowest during heating, and intermediate during isothermal operation, (Fig. 5). These differences were attributed to gravity effects.

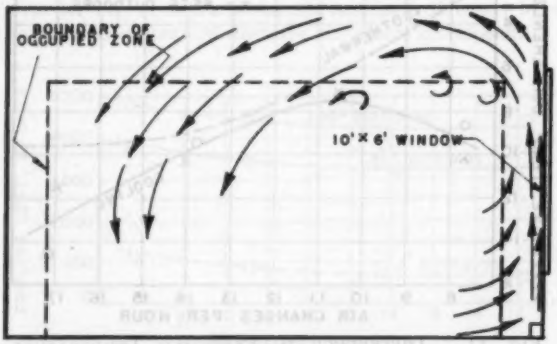

FIG. 13. . . .AIR-FLOW PATTERNS IN TEST ROOM; BASEBOARD DIFFUSERS; 96 F OUTDOORS, 17 AIR CHANGES

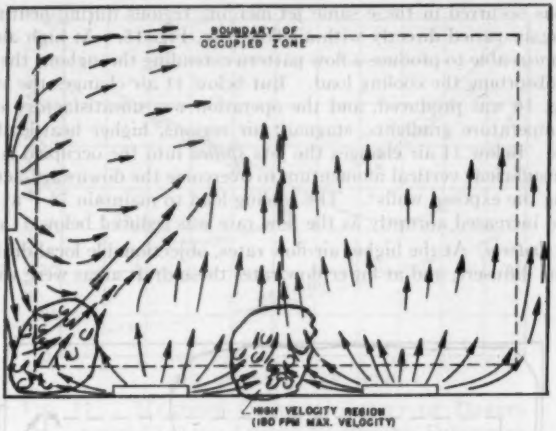

FIG. 14. . . .AIR-FLOW PATTERNS IN TEST ROOM, PLAN VIEW; BASEBOARD DIFFUSERS; 96 F OUTDOORS, 17 AIR CHANGES

The floor-to-ceiling temperature differences were below 3 deg in all tests. With appreciable cooling loads, local drafts were objectionable, and velocities exceeded 50 fpm, (Table 1).

SYSTEM NO. 2, BASEBOARD RADIAL DIFFUSERS

Flow Pattern, Drafts and Cooling Load: The room air-flow pattern shown in Figs. 13 and 14 prevailed when the air volume was above 11 air changes per hr. The plan view of Fig. 14 indicates 2 critical draft regions at the boundaries of the occupied zone, produced by the merging jets from the baseboard diffusers. The critical

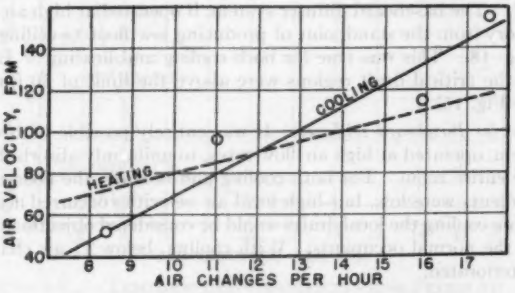

FIG. 15. . . .MAXIMUM AIR VELOCITIES IN CRITICAL DRAFT REGIONS WITH BASEBOARD RADIAL DIFFUSERS

draft regions occurred in these same jet-merging regions during heating[1]. Draft velocities again varied directly with air-flow rate, Fig. 15. At high air-flow rates this system *was* able to produce a flow pattern extending throughout the room, thus uniformly absorbing the cooling load. But below 11 air changes the vertical pattern of Fig. 16 was produced, and the operation was unsatisfactory due to high vertical temperature gradients, stagnant air regions, higher heat-load gains and spot drafts. Below 11 air changes the jets *spilled* into the occupied zone because they had insufficient vertical momentum to overcome the downward-acting gravity forces along the exposed walls*. The cooling load to maintain 76 F at the control station also increased abruptly as the flow rate was reduced below 11 air changes.

Comfort Criteria: At the higher air-flow rates, objectionable local drafts occurred between the diffusers, and at lower flow rates these draft areas were widened. At

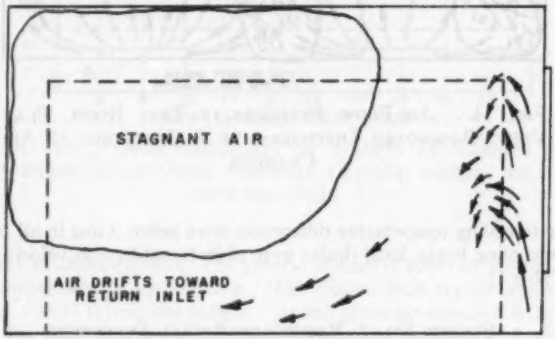

FIG. 16. . . .AIR-FLOW PATTERNS IN TEST ROOM; BASEBOARD DIFFUSERS, 8 AIR CHANGES; 74 F OUTDOORS; SHOWING *Spilling* OF COOL AIR INTO OCCUPIED ZONE

least one-third of the occupants would find the drafts objectionable at these locations, Fig. 17. The baseboard diffuser system, if operated at high air-change rates, was satisfactory from the standpoint of producing low floor-to-ceiling temperature gradients, Fig. 18. This was true for both cooling and heating[1]. In general, the velocities in the critical draft regions were above the limit of 50 fpm specified in Table 1, (see Fig. 15).

Conclusions for Baseboard Diffusers: It was entirely possible with the baseboard diffuser system, operated at high air-flow rates, to uniformly absorb the load and to ventilate the entire room. For both cooling and heating, the floor-to-ceiling temperature gradients were low, but high local air velocities occurred near the exposed walls. During cooling the local drafts would be considered objectionable by at least one-third of the normal occupants. With cooling, below 11 air changes, the performance deteriorated.

* Reference 7 covers the general theory of vertical non-isothermal air jets and how to compute maximum travel of upward projected chilled air or downward projected heated air.

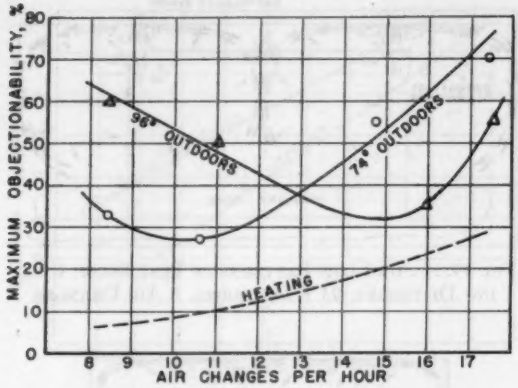

FIG. 17. . . .MAXIMUM OBJECTIONABILITY OF DRAFTS AT VARIOUS COOLING LOADS; BASEBOARD DIFFUSERS

SYSTEM NO. 3, ROUND CEILING DIFFUSERS

Flow Pattern, Drafts and Cooling Load: With 2 commercial ceiling diffusers of the projecting cone type, (6-in. neck), each mounted centrally in one-half of the ceiling, the vertical patterns of Figs. 19 and 20 were produced. In contrast with the operation of the baseboard diffusers, there was no critical flow rate for the ceiling diffusers at which the stagnant zone disappeared. As the flow rate was increased, the air streams traveled farther. The entrainment action also tends to pull the air from the floor level upward, Fig. 21.

Figs. 19 and 20 show that a major quantity of the supply air travels along the ceiling, radially outward from the diffusers, then down the walls, picking up the cooling load near the exposed surfaces. The unoccupied zones near the ceiling and

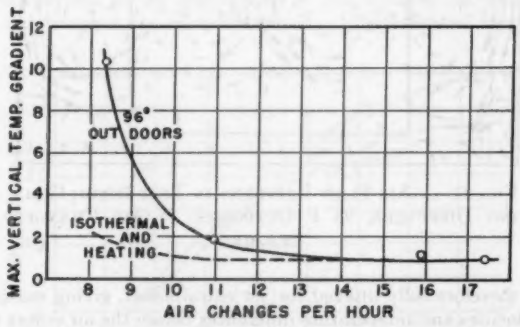

FIG. 18. . . .TEMPERATURE GRADIENTS FROM FLOOR TO CEILING OF OCCUPIED ZONE; BASEBOARD RADIAL DIFFUSERS

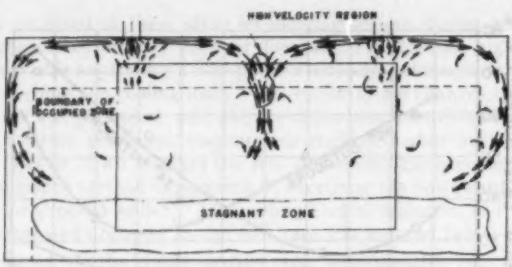

FIG. 19. . . .AIR-FLOW PATTERNS IN TEST ROOM; CEILING DIFFUSERS, 93 F OUTDOORS, 8 AIR CHANGES

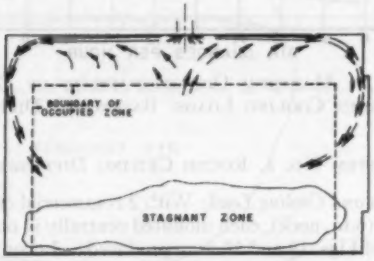

FIG. 20. . . .AIR-FLOW PATTERNS IN TEST ROOM; CEILING DIFFUSERS, 93 F OUTDOORS, 8 AIR CHANGES

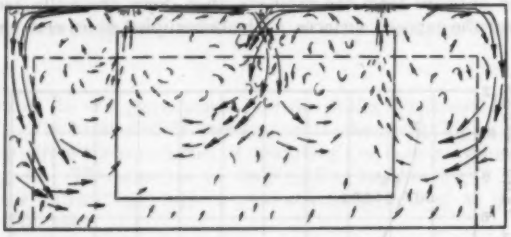

FIG. 21. . . .AIR-FLOW PATTERNS IN TEST ROOM; CEILING DIFFUSERS, 93 F OUTDOORS, 16 AIR CHANGES PER HR

the walls are therefore fully utilized for jet entrainment, giving maximum reduction of jet velocities and temperature differences before the air enters the occupied zone. Air supply temperatures required for 96 F outdoors are shown in Fig. 10.

The critical draft region was between the 2 diffusers, near the center of the room where the air enters the top of the occupied zone, as shown in Fig. 19. The draft

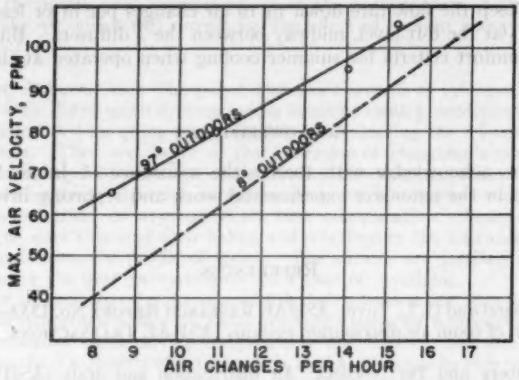

FIG. 22. . . .MAXIMUM DRAFT VELOCITIES IN OCCUPIED ZONE; CEILING DIFFUSERS, COOLING AND HEATING

velocities were linear with air change, Fig. 22. It was therefore concluded that inertia forces again predominated, as in heating. This is in contrast with Fig. 5, which indicates a leveling out of air velocities in the occupied zone due to the buoyancy at low air-change rates and correspondingly low air-supply temperatures.

Comfort Criteria: Although local velocities are slightly high, Fig. 22, the drafts would not be considered objectionable by more than 10 percent of normal occupants if a low flow rate is used, Fig. 23. At flow rates above 10 air changes, however, the drafts become objectionable to a greater number, and the local air velocities are considerably above the prescribed limit of 50 fpm. The floor-to-ceiling temperature gradients in the occupied zone were negligible for all conditions of cooling operation, but they were as much as 15 deg at low-flow rates during heating[1].

Conclusions for Ceiling Diffusers: The air-flow pattern was independent of flow rate, and was excellent for air distribution in that it made full use of unoccupied zone for air entrainment and load absorption. Limitations of room size made it

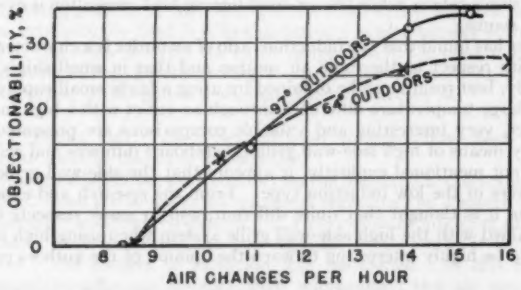

FIG. 23. . . .OBJECTIONABILITY OF DRAFTS AT VARIOUS LOADS; CEILING DIFFUSERS

necessary to keep the flow rate down to 10 air changes per hr or less in order to prevent drafts at the 6-ft level, midway between the 2 diffusers. But the system satisfied all comfort criteria for summer cooling when operated at the lower flow rates.

ACKNOWLEDGMENT

The authors acknowledge with thanks the assistance of James Wahlers and Richard Gido in the extensive experimental work and reporting involved in this project.

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DISCUSSION

G. A. BAKKER, The Hague, Holland (WRITTEN): As the oldest specialized firm for air conditioning in Holland, the field of ship's air conditioning is more important for us than land installations because of the mild climate in Holland.

In ship's air conditioning the problem of room air distribution is encountered, and may be more critical than in residential air-conditioning systems, because of the smaller dimensions of ship's cabins, while the air-conditioning load very often is even larger than for residential rooms.

The discusser has found that the induction ratio of an outlet is a characteristic of major importance with respect to the room air motion and that in small ship's cabins, or in rooms generally, best results can be obtained by using a fairly small supply-air-quantity with a rather large temperature difference through an outlet with a high induction ratio.

In the paper, very interesting and valuable comparisons are presented between air distribution by means of high side-wall grilles, baseboard diffusers and ceiling diffusers, but although not mentioned explicitly, it appears that the side-wall grilles used in the experiments were of the low induction type. From the research and experience of the discusser's firm it is thought that quite different, and in many respects better, results might be obtained with the high side-wall grille system when using high induction outlets. It would be highly interesting to learn the opinion of the authors concerning this subject.

R. W. MCKINLEY, Pittsburgh, Pa.: In most actual rooms, there is an asymmetric radiant energy distribution. In our experience, the distribution of radiant energy

density has frequently been overlooked. Can the authors state if there is a way of introducing the air which will correct for the effect of radiant energy density in the occupied zone?

T. C. MIN, Auburn, Ala.: The paper contains a wealth of information on the performance of 3 air distribution systems under summer cooling conditions. The authors are to be commended for going a step farther by evaluating the 3 systems in terms of human comfort. They are aware of the limitation of Houghten's correlation. The correlation, made some 20 years ago, of air velocity, effective temperature difference, and percentage of occupants objecting to drafts, does not take factors such as radiational condition (symmetrical or asymmetrical) into consideration. Besides, the type of clothing people wore then and their habits and reactions to the tolerance to draft could also be different from the people of today. The authors are justified in comparing 3 systems by using the best correlation on draft thus far available.

However, it is urged that the authors publish their complete test data of the 3 systems, *e.g.* supply air temperature (dry-bulb and wet-bulb), room air temperature distribution, velocity distribution, room surface temperatures and finishes, and illuminating condition, if any, in the TRANSACTIONS as Appendix so that one may be able to make a refinement on the evaluation of the 3 systems in the future when a complete and definitive comfort criteria is established. The Research Laboratory of the Society has undertaken a long-range program of environmental research since 1957, and it is expected results on the relation between human comfort and all environmental factors will be forthcoming. A re-evaluation of the 3 systems is possible at that time only if such worthwhile air distribution test data are recorded with the paper.

In Fig. 1 in the paper there is an extrapolation of comfort data. Could the author elaborate about the test conditions and number of people on which this comfort chart is based? Also, did the heating and cooling conditions and occupancy used for the chart justify its use by the authors as a criterion of comfort?

AUTHORS' CLOSURE (Mr. Reinmann): Mr. McKinley's question about correcting for the effect of radiation energy density is very significant. In this author's own experience and opinion the only way this could be done, if the room air were still, would be to put radiant heaters on the walls, say on the windows, and put radiant type heaters outside. However, this was not done and consequently no comment is possible about that condition as far as extrapolating the data is concerned. One cannot expect that the inside wall temperature would change. At high air-flow rates, the inside wall temperatures probably do not affect the air-flow pattern greatly, since normally there is no radiation on the ceiling. The air-flow pattern might be distorted a little along the windows, if the windows were not too high and not too much radiation had to be absorbed, but this might not be the case.

Concerning the questions of Mr. Min about extrapolation and testing details, these comfort data were actually used first in evaluating a winter heating system as discussed in Reference 1. In order to cover the entire range used in the present study, the extrapolations indicated in Fig. 1 were necessary. As to the test methods and subjects, the details will be found in Houghten's original paper, Reference 4.

AUTHORS' CLOSURE (Mr. Tuve): Several good questions have been raised, but yes or no answers to most of them are not possible.

There is always a conflict between comfort on the one hand and economy on the other. One must admit there are unanswered questions regarding the comfort criteria, but the authors considered the Houghten data of Fig. 1 as reliable as anything available. Some members present may remember the thoroughness of his work; it is reported in the TRANSACTIONS. The subject of radiation was mentioned. In this paper the only measured quantities affecting comfort were temperature and air motion. The test room was arranged to simulate a well-insulated structure with no direct exposure to solar radiation, so that radiation effects were very small.

One condition that deserves more attention when evaluating comfort is the so-called *stagnant zone.* The exact nature of this zone and its effect on comfort have not been determined, except to note that the velocities are so low they can hardly be measured. Is the presence of such a zone to be taken as an indication that the room is not well ventilated, that there is insufficient air mixing? Certainly the problem of mixing conditioned air with room air is more difficult in a room with an 8-ft ceiling than in a high-ceiling room.

In this connection Mr. Bakker's question about the effect of high induction ratio is most pertinent. He apparently refers to the high induction ratio of perforated panel outlets in which the holes are spaced far enough apart so that the small jets act more or less independently. The high entrainment or induction of such an outlet does mean shorter throw and greater mixing with the room air near the outlet. It is known that this is one of the important results accomplished by ceiling outlets and other types of wide-spread diffusers. This high induction permits greater temperature differences between the supply air and the room. But it also uses up the kinetic energy of the original jet so that it is not available for spreading the air into the rest of the room. Is this undesirable? We do not know because the real nature of the stagnant zone has not been determined. It is known that when the active mixing is confined to a small, high induction area and the throw is short, the stagnant zones are likely to appear, as shown in several figures in this paper and in the extensive studies presented in the University of Illinois reports by Konzo, Straub and Gilman.

The studies reported in the present paper were actually undertaken with a different purpose in view, *viz.*, getting the best economy from a heat pump for heating and a refrigeration machine for cooling. In other words special attention was paid to high air volume and low temperature differences, (using high evaporator temperatures in summer and low condenser temperatures in winter). The criterion of success was comfort in terms of air motion and temperature, and naturally it was found that there was a best rate of air change for each system, meaning that there is a limit to either air volume or temperature difference.

Work is still being continued at Case Institute in cooperation with the Society, in an effort to answer some of the remaining questions. Tests with full-scale rooms are just too expensive and time consuming, so quarter-scale and half-scale models are now being used. By the use of tracer techniques it has been found that the stagnant zones are not as stagnant as they were thought to be, in fact they show active mixing. But what about comfort in these zones? It is hoped that the Research Laboratory of the Society will eventually be able to tell more about human reactions as related to air purity or air change in those parts of the room where the temperature is comfortable but where velocity instruments call the air *stagnant.*

1698

No. 1698

SURFACE ODOR ADSORPTION AND RETENTION PROPERTIES OF SURFACES

By W. F. HOPPER*, CLEVELAND, OHIO

This paper is the result of research carried out by the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio

THE PURPOSE of this study was to develop a suitable method by which the odor adsorption and retention characteristics of various surfaces could be measured. To demonstrate the suitability of the method, the odor retention characteristics of several fabrics were measured. Cotton, wool, nylon and rayon were chosen as the test fabrics, and iso-amyl acetate was used as the odor agent. Data on the odor retention properties of the first three of these fabrics are included here. Data on rayon are not presented because the rayon samples did not adsorb enough odor to be measured.

The method developed and explained is essentially an application of the syringe technique[1] of odor evaluation for determining the retention of odors on various surfaces which have been exposed to odorous atmospheres. All steps in the procedure were carried out under controlled environmental conditions.

TEST APPARATUS

Test Rooms: Preparation of samples and odor appraisals were made in 2 test rooms[2] located in the main Laboratory of the ASHRAE Research Building. The rooms, which are separated by a corridor, measured 8 × 12 × 8 ft high and were lined with porcelain-enameled steel having low odor-adsorption qualities. Test room doors were weatherstripped to reduce infiltration of outside air. Inlet air was filtered and deodorized by passing through one bank of mechanical filters, an elec-

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* Research Engineer, ASHRAE Research Laboratory.

[1] Exponent numerals refer to References.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, Lake Placid, N. Y., June 1959.

trostatic precipitator, a second bank of filters, two banks of activated carbon cells, and finally, through a high-resistance, high-efficiency filter. Air from the rooms was not recirculated but exhausted to outdoors. Three separate air-conditioning systems, each with the necessary equipment for cooling, heating, and humidification, served the rooms and the corridor. Air flow was regulated by adjustable dampers on the inlet side of the supply fans and metering orifices were used to measure flow rates.

Odor Adsorption Apparatus: Fabric samples were prepared for test by hanging in a glass jar, 11 in. in diam and 2 ft high, which was fitted with a tight cover. An open dish of odorant was placed in the bottom of the jar, and a second dish of desic-

FIG. 1. . . .EQUIPMENT USED IN SYRINGE METHOD OF ODOR EVALUATION

cant solution was provided to maintain the relative humidity at the desired level. A small fan circulated the air within the jar.

Bottles: Throughout the program, glass bottles ½-gal in size were used for storing and testing the odorized fabric samples. The bottles were large enough to permit several openings before enough dilution took place to alter test results. Odor transfer to or from the corks was eliminated by wrapping the corks in frequently replaced aluminum foil.

Bottle Cleaning Apparatus: Noxious bottle odors were a major problem which appeared during the early stages of the test program. These odors had to be eliminated because they tended to mask or counteract the odor which had been intentionally introduced into the bottles. To reduce the bottle odors to a harmless level, the bottles were washed with dairy brushes, using an effective odorless detergent, and were boiled for 5 min. After boiling, the bottles were placed neck down on a rack, and air which had been filtered and passed through activated carbon, was blown into them.

Syringes and Accessories: For the syringe method of odor evaluation, twelve 100-ml syringes, 12 syringe caps, one 2-ml syringe, one 10-ml syringe and 1 transfer

needle were used in conjunction with an individual nozzle for each panel member. The transfer needle consisted of a needle with a mating head on each end to permit the joining of 2 syringes. The extra mating head was silver soldered in place.

The individual nozzle consisted of a Luer Lok fitting for an 18-gage needle from which the needle had been removed and to which a plastic tube had been attached. The syringes and accessories are shown in Fig. 1.

Washing of the syringes followed the same procedure that was used on the bottles.

DESCRIPTION OF TEST FABRICS

The fabrics chosen for test were matched as closely as possible. The factors considered were weave, thread count, finish and weight per square yard. Since the fabrics were not custom made, but were obtained through standard sources, there were minor differences in all the factors with the exception of the weave. Table 1 lists the properties of the fabrics used in this study.

TABLE 1. . . .PROPERTIES OF FABRICS TESTED

FIBER	FINISH	WEIGHT— OZ/SQ YD	WEAVE	THREAD COUNT
Cotton	Greige	7.30	Plain	48 x 48
Wool	Scoured & Fulled	7.5 to 8.1	Plain	35 x 41
Nylon	Greige	8.0	Plain	38 x 38
Rayon	Greige	7.60	Plain	38 x 38

Odorants: Hexanoic acid was the first odorant used in the testing. This odorant, which may be best described as *goaty*, was not only unpleasant to smell, but reacted with the metallic parts in the odorizing jar. For these reasons, its use was discontinued.

To replace hexanoic acid, isoamyl acetate was chosen because it had a strong and not unpleasant banana-like odor, and was not corrosive. Since the odor of iso-amyl acetate is extremely perceptible, it was necessary to dilute the odorant so that better control of the samples in the loading tank could be obtained. Benzyl benzoate, a nearly odorless diluent, was used to reduce the vapor pressure of the odorant to 10 percent of its original level. The mixture used was 13 parts by weight of iso-amyl acetate with 191 parts by weight of benzyl benzoate. With this dilution at 75 F and 50 percent relative humidity, the odor level of the iso-amyl acetate in the loading tank, in terms of threshold units, was reduced to one million.

The Threshold Concept: In odor work, the term threshold is used as a measure of the concentration of an odorous mixture. The threshold level of odor perception is the lowest concentration of odorous air mixtures which, when mixed with odor-free air, can be consistently identified. If an odorous mixture such as the one in the loading tank has to be diluted to 1 ppm to reach a threshold level, then the original odorous air is one million times more concentrated than the threshold-level mixture or it has an odor level of one million thresholds.

Syringes loaded with odor-laden air were prepared for classification as to odor level by a dilution process using odor-free air. By varying the amounts of odor-

free air added to the odor sample under investigation, it was possible to adjust the odor concentration in the syringes until the odor was just recognizable or until the recognition threshold level was reached. For example, with several syringes loaded from the same odor source, one diluted with 9 parts of air to 1 part of odor-laden air would impress on an observer a lower sensation of odor level than would a similar syringe loaded with 3 parts of odor-free air and 1 part of odor-laden air. Consider, for example, the contents of the 100-ml reference syringe used in the investigation. At the 9 dilution loading, it would contain 10 ml of odor-laden air and 90 ml of odor-free air, while at a dilution of 3 it would contain 25 ml of odor-laden air and 75 ml of odor-free air. If the odorant in the syringe with the 9 dilution loading was

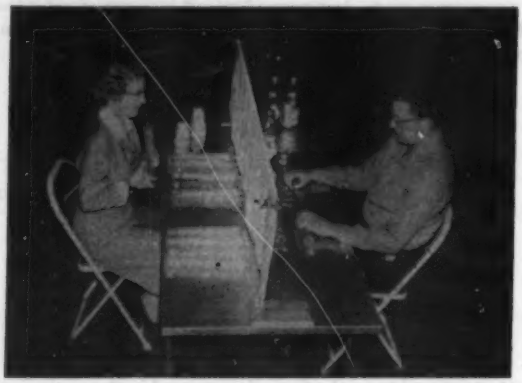

FIG. 2. . . .ODOR EVALUATION TEST BOTTLES AND SYRINGES

just perceptible, the undiluted odorant would have a threshold level of 10, 9-dilution plus one. If, however, the original odor were weaker and only 3 parts of air were needed to dilute it to threshold level, it would be described as a 4-threshold odor, namely, 3 dilutions plus one. For a much stronger odor, the ratio might be equivalent to 0.1 ml of odor-laden air and 99.9 ml of odor-free air or a ratio of 1 to 999; and this mixture, if its odor is just detectable, would be described as having a threshold level of 1000 or merely as being at 1000 thresholds.

PREPARATION OF TEST SAMPLES

Cloth samples, used early in the program for the bottle testing method were of various sizes, ranging from 6 × 6 in. to ¾ × ¾ in., each having an area one-half of the next sample. All samples for the syringe method of evaluation were 6 in. square. Cloth samples were odorized for 24 hr in the loading tank and then aired at the desired environmental conditions for various periods of time ranging from 0 to 24 hr. At the end of the airing period, the samples were stored in tightly corked bottles for at least 24 hr prior to making odor evaluations. The 24-hr periods for

odor loading and storage before testing were adhered to—to insure equilibrium conditions.

ODOR EVALUATION PROCEDURE

Bottle Testing Method: At the start of the program, bottle testing was suggested as a method which might be used to determine threshold levels. The bottle method consists of setting up different odor levels in a group of bottles by placing odorous fabric samples of different sizes in them. By removing the corks one at a time and sniffing, each subject selected a bottle in which the odor was at his threshold level.

The method showed numerous inconsistencies which could have been caused by a number of reasons, an obvious one being the possibility of cross-currents of air

TEST: 12 N DATE: MAY 22 1958 MATERIAL: NYLON
ODORANT: ISO AMYL ACETATE
LOADING TIME: 24 HOURS LOADING CONDITIONS D.B. 75F R.H. 50%
AIRING TIME: 240 MINUTES AIRING CONDITIONS D.B. 75F R.H. 50%
EVALUATION CONDITIONS D.B. 75F R.H. 50%

PERCEIVE / NAME	YES	NO	YES	NO	YES	NO	YES	NO	THRESHOLDS	SCORE THIS SHEET
EMC		✓		✓	✓			✓	33	AVERAGE 50
LFS		✓	✓		✓			✓	50	
WFK		✓		✓	✓			✓	33	MEDIAN 41
IMJ		✓		✓	✓			✓	33	
MF	✓		✓		✓			✓	100	WEIGHTED GEOMETRIC MEAN 46
WFH		✓	✓		✓			✓	50	
SYRINGE LOADING	99-1		49-1		24-1		BACK-GROUND			

FIG. 3. . . .SAMPLE FABRIC ODOR DATA SHEET

mixing with the bottle odors at the mouth of the bottle during the process of test sniffing. The syringe testing method was so much more reliable that it was decided to discontinue bottle testing completely.

Syringe Testing Technique: To evaluate the odorous mixture in the bottle, a syringe technique adopted by ASTM[1] was used. The bottle cork was replaced by the cork and tube combination shown in the bottle in Fig. 1. The tube had a Luer Lok fitting on the outer end. Odorant mixture was then withdrawn from the bottle to fill a 100-ml syringe. This syringe became the storage syringe, and the odorant mixture it contained was used as needed for testing. Transfer of the odorant mixture from the storage syringe was accomplished by use of a transfer needle, also shown in Fig. 1. This was connected to a 2-ml or 10-ml transfer syringe, the size depending on the quantity needed. The transfer syringe, after loading, was then attached by the transfer needle to other 100-ml syringes which were prepared for testing. Each test syringe was charged by introducing a predetermined quantity of odorant mixture and then filling to the 100-ml mark with air from the relatively odor-free test room in which the odor evaluations were made. The syringe was then capped and allowed to stand for at least 15 sec to permit

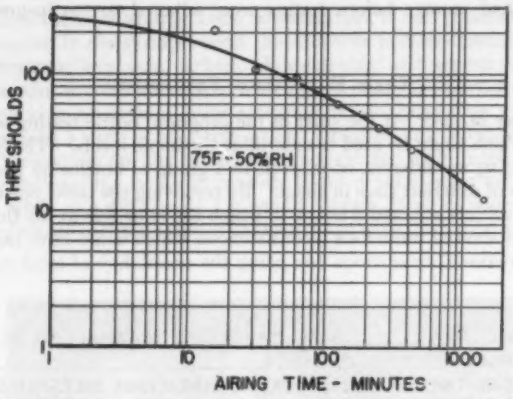

FIG. 4. . . .ODOR RETENTION CURVE FOR COTTON FABRIC TESTED

thorough diffusion of the odorant. At the end of this period, a syringe could either be used for a test or its contents could be further diluted. When further dilution was desired, part of the contents were pumped out, the syringe was refilled with test-room air, and after 15 sec, was ready for use.

For each panel member, 3 test syringes, each containing a different concentration of odorant, were prepared for each test sample. These syringes, together with a fourth syringe containing background odor, were arranged in random order for

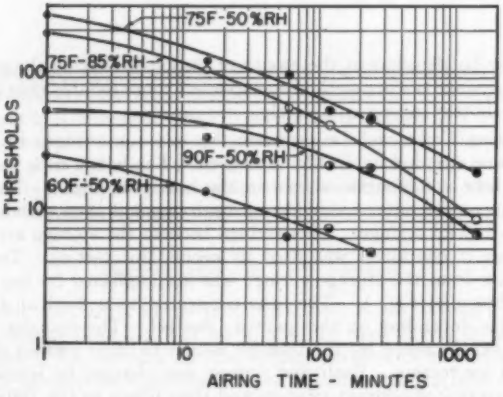

FIG. 5. . . .ODOR RETENTION CURVES FOR NYLON FABRIC TESTED

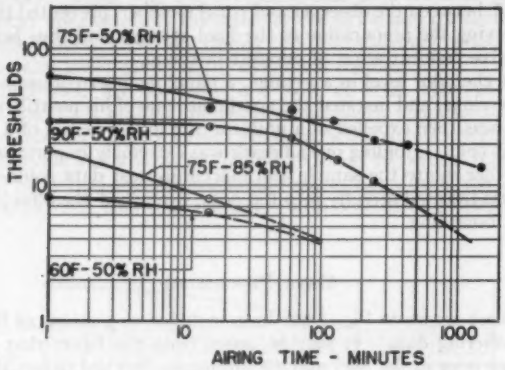

FIG. 6. . . .ODOR RETENTION CURVES FOR WOOL FABRIC TESTED

examination by the panel member. Two such sets of syringes could be examined by a panel member at one sitting before olfactory fatigue became apparent. In addition to the 2 sets of 4 syringes, an additional syringe containing known background odor was always available to the panel member for reference.

After a panel member had sniffed and reported his reactions to the 4 syringes in a given set, the observer prepared additional syringes containing concentrations slightly lower than the lowest concentration originally detected. In this way the threshold level for each panel member was determined more closely. A low screen,

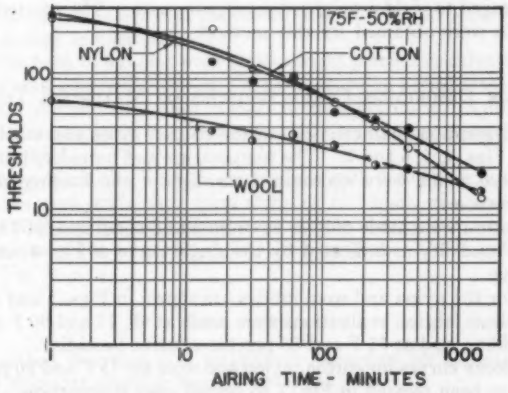

FIG. 7. . . .ODOR RETENTION CURVES FOR COTTON, NYLON AND WOOL

shown in Fig. 2, between the observer and panel member, prevented the panel member from observing the preparation of the final syringes and thus being influenced as to the relative concentration of odorant in them.

To evaluate the odor level in a syringe, a panel member connected his personal nozzle to the syringe, and inserted the nozzle into one of his nostrils. The mixture in the syringe was then expelled at a uniform rate over a period of 2 to 3 sec while the breath was held. Holding one's breath was extremely important since breathing would further dilute the sample and inaccurate test data could result. Since the odorant was injected directly into the nose, it reached the olfactory bulb without the aid of breathing.

DATA PROCESSING

The data sheet shown in Fig. 3 has been included as a means of illustrating the method of gathering data. It may be noted from the figure that subject EMC could detect no odor in the 99-1 and 49-1 dilutions, but did detect the odor in the 24-1 dilution. Thus the dilution ratio which would be at threshold level for this subject was between 24-1 and 49-1. Additional syringes were prepared having dilutions within this range, and a 32-1 dilution was finally judged to be at threshold level, or as noted in the threshold column, EMC's odorant level was 33 thresholds. The dilutions used in the final syringes to pin-point the odor intensity were not noted on the data sheet.

Analyses of the data presented a problem since there were large variations in perception levels among panelists. Averaging the results overemphasized the large variations among panelists, and the resulting average value was open to question. A median method gave less scatter of points but was only moderately successful. A third method, the weighted geometic mean, gave the most consistent record of the test data. The weighted geometric mean (for one airing period) was found by plotting each panelist's perception level at that airing period on logarithmic paper. After the points were plotted, the linear distance from the abscissa to each point was measured in length units, centimeters or inches. The total of these linear distances was then averaged to yield a weighted geometric mean. The plotted points on the several graphs were obtained by this method.

ODOR DESORPTION CHARACTERISTICS OF MATERIALS TESTED

The odor desorption characteristics of the cotton, nylon and wool fabrics tested are shown in Figs. 4, 5, 6 and 7. The temperature and humidity indicated on each curve are those which were maintained throughout the loading, desorption and evaluation processes.

Tests on cotton were made only at environmental conditions of 75 F and 50 percent relative humidity as indicated by the single curve of Fig. 4; other tests were more extensive.

The data for the nylon and wool fabrics are shown in Figs. 5 and 6 respectively. For both of these fabrics, evaluations were made at 60, 75 and 90 F and 50 percent relative humidity, and at 75 F and 85 percent relative humidity.

The odor decay curves for cotton, nylon and wool for 75 F and 50 percent relative humidity have been plotted in Fig. 7 to permit easy comparison. The threshold level at 1-min airing time may be taken as a measure of the odor pickup of each fabric during the 24-hr charging period, while the slope of the curve indicates the

rate of odor decay. From Fig. 7 it is evident that both the odor pickup and the rate of odor decay are essentially the same for the nylon and cotton samples tested. The wool sample adsorbed less odor during the 24-hr charging period but its rate of odor decay was also lower than that of nylon and cotton.

The data presented are for only one bolt of each type of fabric, for a single odorant, and for a limited number of atmospheric conditions. Although no general conclusions can be drawn from the data, some trends are suggested. The curves for both nylon and wool at 60, 75 and 90 F and 50 percent relative humidity (Figs. 5 and 6) show that the maximum odor adsorption during the 24-hr charging period occurred at 75 F, and that the adsorption both at 60 F and 90 F was con-

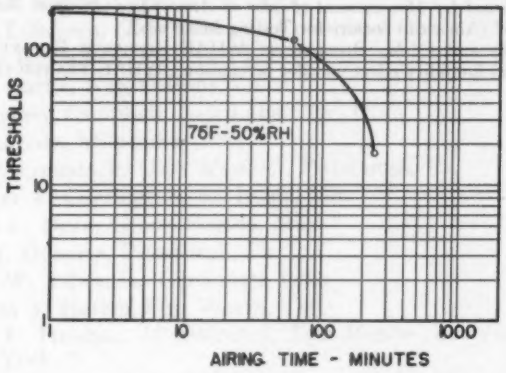

FIG. 8. . . .ODOR RETENTION CURVE FOR WOOD

siderably lower. This implies that maximum odor adsorption occurs on fabrics at about normal indoor temperature.

The curves for both nylon and wool at 50 and 85 percent relative humidity and 75 F show that odor adsorption during the 24-hr charging period was greater at the lower humidity. Additional studies on more types of fabrics with a number of odorants and at other atmospheric conditions are needed to determine whether the trends suggested are typical.

For the purpose of making a rough comparison with a completely different type of surface, 3 tests were made with wood strips of fir 1½-in. wide and ⅛-in. thick. The total area of the strips comprising each sample was approximately 72 sq in., the same as the area of the two sides of the fabric samples. The data for these tests are plotted in Fig. 8.

CONCLUSIONS

1. The test method described is a satisfactory procedure for evaluating the odor adsorption and retention characteristics of many surfaces.
2. The odor adsorption by the rayon sample tested was so low that it could not be detected by the test method used in this program.

3. The syringe technique for odor evaluation is more accurate than the bottle method which was used in the earlier part of the program.

4. The data presented were gathered in the process of determining the suitability of the test method, and although limited in scope, do suggest the following trends which may be worthy of note:

(*a*) At 50 percent relative humidity, maximum odor adsorption during a 24-hr period occurs at about 75 F, and is considerably less at 60 F and 90 F.

(*b*) At 75 F, odor adsorption during a 24-hr period is greater at 50 percent than at 85 percent relative humidity.

Further work will be necessary to confirm these preliminary findings.

REFERENCES

[1] *Standard Method for Measurement of Odor in Atmosphere (Dilution Method): Designation* D 1391-57 (American Society for Testing Materials).

[2] W. F. Kerka and C. M. Humphreys: ASHAE RESEARCH REPORT No. 1587—Temperature and humidity effect on odor perception (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 531).

In Memoriam 1959

Name	Joined
Ray W. Adam, Detroit, Mich.	1937
Eugene E. Adams, Jackson Heights, N. Y.	1938
Joseph M. Aikman, Clearwater, Fla.	1959
George R. Anderson, Fort Wayne, Ind.	1958
Harry H. Angus, (*Life Member*), Toronto, Ont., Canada	1918
William E. Beggs, (*Life Member*), Seattle, Wash.	1927
John G. Bellas, Springfield, Mass.	1957
Myron C. Beman, (*Life Member*), Buffalo, N. Y.	1925
George I. Boone, Great Neck, N. Y.	1937
Aubrey I. Brown, (*Life Member*), Columbus, Ohio	1923
William H. Bunten, Melrose Park, Pa.	1953
John S. Burke, New Orleans, La.	1942
G. Courtney Campbell, Salt Lake City, Utah	1945
T. R. Christy, Milwaukee, Wis.	1950
Glen M. Comstock, (*Life Member*), Pittsburgh, Pa.	1926
Mortimer F. Corrington, St. Louis, Mo.	1952
Vernon C. Dean, Grand Rapids, Mich.	1945
Roger J. Doherty, Pittsburgh, Pa.	1950
Arthur W. Edwards, Cincinnati, Ohio	1936
Benjamin J. Fjeseth, Fort Worth, Tex.	1955
Walter L. Fleisher, (*Presidential, Life Member, and Fellow*), New York, N. Y.	1914
James A. Garland, Cambridge, Mass.	1958
Leo M. Gorsey,* Burbank, Calif.	1955
Edward C. Haas, Johnstown, Pa.	1957
Thomas J. Halligan,* Savannah, Ga.	1957
Robert Hockman, Cincinnati, Ohio	1944
George A. Horne, (*Honorary, Presidential, Life Member, and Fellow*), Lanoka Harbor, N. J.	1911
Evan A. Joy, Pacoima, Calif.	1955
Charles F. Kettering, (*Honorary Member*), Dayton, Ohio	1956
Herman C. Koenig, New York, N. Y.	1952
Hans H. Kohlenberger, Fullerton, Calif.	1956
Harry E. Kottcamp,* York, Pa.	1945
W. H. Knowles, Jersey City, N. J.	1943
Conrad Kramer, Pawtucket, R. I.	1952
William H. Krapohl, Boston, Mass.	1941
Robert M. Krohmer, Marion, Ohio	1955
Edward F. Lankes,* Canonsburg, Pa.	1955

* Died in 1958, but notification received in 1959.

In Memoriam 1959 (*Continued*)

Name	Joined
Donald A. Lassus, New Orleans, La.	1955
Victor M. Lathers, Rockford, Ill.	1945
Harry E. Lewis, Chicago, Ill.	1939
Harold A. Lockhart, Morton Grove, Ill.	1935
James S. Malone, St. Louis, Mo.	1936
Walter N. Mann, Redhill, Surrey, England	1939
Leon T. Mart, Kansas City, Mo.	1941
Ivan McDonald, Toronto, Ont., Canada	1938
Herbert W. Meinholtz, Tulsa, Okla.	1936
Frank A. Merrill, Marblehead, Mass.	1934
Robert A. Miller, Tarentum, Pa.	1931
Daniel J. Montague, Coatesville, Pa.	1944
John L. Morris, Williamson, W. Va.	1957
Chester B. Morrison, (*Life Member*), Scarsdale, N. Y.	1920
Louis L. Narowetz, (*Life Member*), Wilmette, Ill.	1912
William H. Neekamp, Glastonbury, Conn.	1955
John E. Nicholas, (*Fellow*), State College, Pa.	1930
Gustav E. Olsen,* Arverne, N. Y.	1929
Ralph S. Parvis, Newark, Del.	1938
Neil H. Peterson, San Francisco, Calif.	1937
Bernard E. Proctor, Cambridge, Mass.	1938
Raymond R. Quimet, Montreal, Que., Canada	1956
T. H. Ray, Jr., Pittsburgh, Pa.	1952
Olan E. Reese, Houston, Tex.	1950
John A. Rohrs, Jr., Denver, Colo.	1958
Edward R. Sabin,* (*Life Member*), Allenhurst, N. J.	1919
Warren H. Stover, Corvallis, Ore.	1946
William H. Sullivan, Greensboro, N. C.	1943
Walter C. Titchener, Wilson, N. C.	1956
John I. Trimble, Columbus, Ohio	1945
W. J. Uhlhorn, (*Life Member*) Broadview, Ill.	1920
Vincent F. Vallero,* Denver, Colo.	1957
Carl J. Vespasian, Peoria, Ill.	1937
Guy A. Voorhees, Monrovia, Ind.	1937
Edmond H. Worland, Cincinnati, Ohio	1959
Dewitt H. Wyatt, Columbus, Ohio	1936
Jack Wright, Montreal, Que., Canada	1944
Edwin G. Zanone, Louisville, Ky.	1953

* Died in 1958, but notification received in 1959.

Walter L. Fleisher

1888–1959

Walter L. Fleisher, a past president of ASHAE, and president of Air & Refrigeration Corporation, New York, N. Y., died on April 18, 1959, as a result of injuries received in an automobile accident. He lived at Saw Mill Farm, New City, N. Y.

Mr. Fleisher was one of the first engineers to design and install air-conditioning equipment in the United States. He had the distinction of having the grade of Fellow conferred upon him by both the ASHAE and ASRE, and was elected to Life Membership in ASHAE in 1945. Then, with the completion of the merger he became a presidential member of ASHRAE, as well as a Life Member and Fellow.

A native Philadelphian, he was graduated from the University of Pennsylvania with a B.S. in 1900 and received an M.S. in 1901. He was awarded an honorary degree in mechanical engineering by the University in 1931.

Mr. Fleisher began his engineering career with his association as engineer-in-chief for Francis Bros. and Jellett, Philadelphia, a position he left in 1911 to form the W. L. Fleisher & Co., Inc., New York, N. Y., which he served as president-treasurer and chief engineer until it was merged with two other companies under the name of The Cooling and Air Conditioning Corporation.

In 1930 he established a consulting office in his own name in New York, offering services in patent and industrial analysis, as well as advisory work. He organized the Air & Refrigeration Corporation in 1934 and was its president until his death.

In addition to serving ASHAE as president, first vice president and second vice president, in 1941, 1940, and 1939, respectively, and as a member of its Council from 1936–42, Mr. Fleisher served on numerous Society committees, either as chairman or as a member. These included the Executive, Finance, Membership, the F. Paul Anderson, War Service, Research Fund Raising, and the History Committee in 1947. As chairman of the Guide Publication Committee in 1933–34, he

introduced several new features and directed the editorial policy of THE GUIDE. He also served on many of the Society's Special Committees.

Always interested in the research activities of the Society, he gave intensive attention to this for many years. In doing so he served on numerous Technical Advisory Committees, and was also a member of the Committee on Research from 1935–40, serving as its chairman during 1938–39.

In 1953 Mr. Fleisher received the F. Paul Anderson Medal in recognition of his outstanding contributions to the advancement of the arts and sciences of heating, ventilating, and air conditioning in the field of human comfort.

Mr. Fleisher was also extremely active in the affairs of the New York Chapter of ASHAE for many years and served as its president 1920–21.

Comfort air conditioning was a subject with which Mr. Fleisher was closely identified from the beginning of his professional career in 1903. In that year he supervised the engineering of the ventilating system for the New Amsterdam Theater in New York. He designed the first theater cooling system employing an air washer for the Fulton Theater in New York. In succeeding years he designed, supervised, or installed systems in the Shubert, Ames, Forty-fourth Street, and Maxine Elliot Theaters in New York, and many others in Chicago, St. Louis, and in Newark, N. J. One of the first applications of refrigeration for cooling a theater was made under his supervision at the Hippodrome in New York in 1905.

Mr. Fleisher's leadership is evidenced by the fact that he held over 100 patents in this and foreign countries in the fields of cooling, ventilating, controlling, cleaning, drying, and processing. He also invented the capillary conditioner and patented the so-called local recirculation system in which the fresh air is intensively conditioned to create a recirculation at the point of discharge.

With Dr. Willis H. Carrier, he gave comfort conditioning its first big development in the air conditioning of the J. L. Hudson store in Detroit in 1925. This marked the beginning of the wide use of the by-pass system of air conditioning for comfort.

Continuing his interest in the field of human comfort he served as chairman of the ASHAE, ASRE, ASME, ARI and Medical Profession Joint Committee for the Revision of the Code of Minimum Requirements for Comfort Air Conditioning from 1950–57.

During World War II, Mr. Fleisher, with Rear Admiral Morgan Watt and Commander T. H. Urdahl, was responsible for important improvements in the habitability of warships.

In addition to his Society activities, he served on the Advisory Council of the Department of Mechanical Engineering at Princeton University; headed the Standards Committee of the HPACCNA when codes were being developed for computing the heating and ventilating requirements of buildings; and served as chairman of the Advisory Committee to the Department of Air Pollution Control of New York City. He also held membership in the American Bakery Engineers; ASME (Life Member); the Heating, Piping & Air Conditioning Contractors National Association (Honorary Member); the National Fire Protection Association, the New York Academy of Sciences, and the Princeton Club.

He is survived by two sons, Walter L., Jr., also a member of the Society, and Jeffrey; a daughter, Mrs. Anna Fleisher Shore; and four sisters.

George A. Horne

1877–1959

George A. Horne, president (1924) and honorary member of the former American Society of Refrigerating Engineers, died in May at the age of 82.

A graduate of Lehigh University, his early work in the field of chemical engineering was in the laboratories of Thomas A. Edison. Later he joined the staff of B. T. Babbitt, soap manufacturers, as chief chemist, where he had complete charge of research and experimental work, developing new formulas and designing a small laboratory vacuum apparatus for the distillation of fatty acids, also a vacuum still for refining glycerin.

As vice president and chief engineer of the Merchants Refrigerating Co., New York, N. Y., he waged a strenuous campaign throughout his career to improve cold storage practices and to develop the field of refrigerated warehousing.

Among the many projects in which he was engaged, was research work on egg storage (with the late Mary E. Pennington). He was a pioneer on research on the application of ozone to egg storage rooms and humidity control.

He was the author of a series of technical papers which he contributed to ASRE, the most noted of which covered compound ammonia compression. His range of subject matter included other subjects of refrigeration design, as well as the engineer's place in society. The title of his presidential address, which he delivered before the Eleventh Western Meeting of the ASRE in Cleveland, Ohio, in May 1924, was "The Place of the Engineer in Modern Civilization."

Among the papers appearing in *Refrigerating Engineering* were a "Theory of Cooling Towers Compared with Results in Practice" (with the late B. H. Coffey); and "Performance of Single Acting Simple Ammonia Compressor, and Tubular Condensers."

Mr. Horne, along with Fred Ophyls, prepared a paper entitled "Recent Improvements in Refrigerating Apparatus," which was presented at the Fourth International Congress of Refrigeration in London, in 1924.

In 1936, he presented his views on "Refrigerated Warehouse Operating Costs" at the Seventh International Congress of Refrigeration in the Hague, at which he was the official United States delegate.

Prior to and following his presidency of ASRE, Mr. Horne was active in the Society's interest. Besides the many ASRE committees on which he served, he was also a member of the ASME main Committee for Power Test Codes, and chairman of the Joint ASME-ASRE subcommittee on Refrigerating Systems.

INDEX

A

Abrasion Testing of Enameled Wires in Gaseous and Liquid Refrigerants, Repeated Scrape, by F. F. TRUNZO, G. J. BICH, and G. W. HEWITT 675
Accountant's Report 53
Acoustical Testing of Mufflers for Halogenated Hydrocarbon and Oil Refrigeration Systems, by A. F. MARTZ, JR., J. L. MARTIN, and W. R. DANIELSON 479
Discussion 487
Actions of the Governing Bodies (ASRE Council and ASHAE Council) 4
Affidavit, and Consent of Education Department 41
Agreement for Consolidation 13
Air-Conditioning System, Free Piston Compressor in an, by R. J. MCCRORY, R. W. KING, and J. H. MCNINCH 377
Air Cross Flow, Frost Formation and Heat Transfer on a Cylinder Surface in Humid:
Part I—Experimental Study 213
Part II—Theoretical Study and Conclusions 229
Air Distribution Systems for Summer Cooling, Evaluation of Three Room, by J. J. REINMANN, ALFRED KOESTEL, and G. L. TUVE 717
ALGREN, A. B., and P. M. CHUNG, Frost Formation and Heat Transfer on a Cylinder Surface in Humid Air Cross Flow:
Part I—Experimental Study 213
Part II—Theoretical Study and Conclusions 229
An Analysis of Turbine Driven Centrifugal Refrigeration Systems, by W. G. DORSEY, JR. 349
Discussion 353
Anemometer, Omni-Directional Low-Speed Hot-Wire, by J. F. KEMP 205
Announcement to All ASHAE Members on the Merger 11
Annual Energy Use for Electric Space Heating, by R. L. BOYD 259
Discussion 266
Annual Meeting, 1959, ASHAE Sixty-fifth (Philadelphia, Pa.) 47
Accountant's Report 53
Awards
ASHAE-Homer Addams 48
F. Paul Anderson Medal 48
1959 Award of Merit (Life Members' Club) 48
By-Law Amendments, ASHAE Proposed 62
Committee on Arrangements 82
Committee on Research—1958, Annual Report of 63
Council, Report of 60
Fellows, Certificate Presentation 48
Inspectors of Election, Report of 74
President, Report of 49
Program 78
Resolutions Committee, Report of 76
Symposiums
Corrosion and Water Treatment 76
Heat Pump Performance 76
Hydronics 75
Treasurer's Report 50
Annual Meeting, 1959, ASHRAE (Lake Placid, New York) 341
Awards
Klixon Award for 1958 342
Wolverine Award for 1958 342
Business Session
President, Report of 341
Meeting Arrangements Committee 348
Program 343
Annual Report of Committee on Research—1958 63

Apple Storage Process and Its Requirements on Refrigeration Structures and Systems, Controlled Atmosphere, by J. W. ZAHRADNIK 589
ASH, R. S., Evaporative Cooling for Common Storages of Fruits and Vegetables .. 571
ASHAE Proposed By-Law Amendments 62
ASHAE Sixty-fifth Annual Meeting, 1959 (Philadelphia, Pennsylvania) 47
ASHLEY, C. M., Discussion on Acoustical Testing of Mufflers for Halogenated Hydrocarbon and Oil Refrigeration Systems 487
Discussion on Annual Energy Use for Electric Space Heating 267
Discussion on Investigation and Control of Refrigerator Noise 468
ASHLEY, C. M., Proposed ASHRAE Standards for the Measurement of Sound from Equipment 601
ASHRAE Annual Meeting, 1959 (Lake Placid, New York) 341
ASHRAE Standards for the Measurement of Sound from Equipment, Proposed, by C. M. ASHLEY 601
Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems, Controlled, by J. W. ZAHRADNIK 589
Automatic Computer for Fan Testing, by C. H. POUNTNEY, JR. and D. W. SKIPWORTH 393
Awnings, Heat Gain through Windows Shaded by Metal, by NECATI OZISIK and L. F. SCHUTRUM 311

B

BAHNFLETH, D. R., and J. R. WRIGHT, Supply Outlet Locations for Basement Heating 137
BAHNFLETH, D. R., Discussion on Omni-Directional Low-Speed Hot-Wire Anemometer 211
BAILLIF, E. A., and J. P. LAUGHLIN, Investigation and Control of Refrigerator Noise 443
BAKKER, G. A., Discussion on Evaluation of Three Room Air Distribution Systems for Summer Cooling 732
Ballot for Officers, 1959 (ASHAE) 74
Ballots Cast, Total Number of (ASHAE and ASRE) 34
Basement Heating, Supply Outlet Locations for, by J. R. WRIGHT and D. R. BAHNFLETH 137
BASTIAN, L. C., Discussion on Investigation and Control of Refrigerator Noise .. 469
BEACH, W. A., R. W. SAGE, and H. F. SCHROEDER, Effect of Fuel Composition on Oil Burner Noise 627
BEDFORD, THOMAS, Discussion on Environment Reactions in the 80 to 105 F Zone. 131
BEDFORD, THOMAS, Research on Heating and Ventilation in Relation to Human Comfort 83
BICH, G. J., G. W. HEWITT, and F. F. TRUNZO, Repeated Scrape Abrasion Testing of Enameled Wires in Gaseous and Liquid Refrigerants 675
BIEHN, G. L., Discussion on Evaluation of Filters for Removing Irritants from Polluted Air 415
BINDER, R. C., Some Methods for Investigating Noise from Compressors Used on Household Refrigerators 471
BOLING, CECIL, Report of President (ASHRAE) 341
BOYD, R. L., Annual Energy Use for Electric Space Heating 259
BOYLAN, W. H., Discussion on Effect of Fuel Composition on Oil Burner Noise .. 633
BROWN, H. C., Discussion on Downward Heat Transfer through a Joist Space ... 245
BROWN, W. G., Discussion on a New Method of Chimney Design and Performance Evaluation 439
BRYAN, W. L., Discussion on an Analysis of Turbine Driven Centrifugal Refrigeration Systems 353
BUCHANAN, R. S., Discussion on Investigation and Control of Refrigerator Noise. 469
BUCHBERG, HARRY, D. LEBELL, and W. B. DRAKE, Load Calculations Using Pretabulated Admittance Functions 515
Transfer Admittance Functions for Typical Composite Wall Sections 523
BUCHBERG, HARRY, Discussion on a New Method for Calculating Radiant Exchanges 330
Discussion on Large-Scale Wall Heat-Flow Measuring Apparatus 550
Discussion on Solar Radiant Gains through Directional Glass Exposure 513

BUEHLER, LEON, JR., Discussion on Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems 596
Bulletins
Corrosion and Water Treatment 76
Cryogenics 343
Heat Pump Performance 76
Hydronics 75
Industrial Ventilation 342
BURKE, J. J., Discussion on an Analysis of Turbine Driven Centrifugal Refrigeration Systems 353
By-Laws, Consolidated Society
Amendments 30
Board of Directors and Officers 22
Chapters and Regions 29
Committees 25
Elections 24
Fees and Annual Dues 21
Funds 30
Government 18
Incorporation 18
Interim Provisions 31
Meetings 24
Membership 19
By-Laws for Proposed Consolidated Society AMERICAN SOCIETY OF HEATING, REFRIGERATING, AND AIR-CONDITIONING ENGINEERS, INC. 18

C

CANOVA, FRED, Discussion on Annual Energy Use for Electric Space Heating 267
CASWELL, H. E., and F. J. WIESNER, JR., Effects of Refrigerant Properties on Centrifugal Compressor Impeller Dimensions and Stage Performance 355
Centrifugal Compressor Impeller Dimensions and Stage Performance, Effects of Refrigerant Properties on, by F. J. WIESNER, JR. and H. E. CASWELL 355
Centrifugal Refrigeration Systems, An Analysis of Turbine Driven, by W. G. DORSEY, JR. 349
Certificate of Consolidation 42
CHADDOCK, J. B., Discussion on Proposed ASHRAE Standards for the Measurement of Sound from Equipment 610
Chimney Design and Performance Evaluation, A New Method of, by W. C. MOFFAT and W. G. COLBORNE 431
Chimney Performance, Fundamental Analysis of, by W. G. COLBORNE and W. C. MOFFATT 417
CHUNG, P. M., and A. B. ALGREN, Frost Formation and Heat Transfer on a Cylinder Surface in Humid Air Cross Flow:
Part I—Experimental Study 213
Part II—Theoretical Study and Conclusions 229
CHUNG, P. M., and C. E. LUND, Downward Heat Transfer through a Joist Space. 245
COLBORNE, W. G., and W. C. MOFFATT, A New Method of Chimney Design and Performance 431
Fundamental Analysis of Chimney Performance 417
Combustion Noise in Oil-Burning Equipment, Low Frequency, by G. J. SANDERS and W. E. LAWRIE 181
Comfort, Research on Heating and Ventilation in Relation to Human, by THOMAS BEDFORD 83
Committee on Arrangements (ASHAE Sixty-fifth Annual Meeting, Philadelphia, Pa.) 82
Committee on Research—1958, Report of 63
Committees, Names of General 25
Comparison of Fluoroalkane Absorption Refrigerants, by B. J. EISEMAN, JR. 635
Discussion 649
Compressor Impeller Dimensions and Stage Performance, Effects of Refrigerant Properties on Centrifugal, by F. J. WIESNER, JR. and H. E. CASWELL 355
Compressor in an Air-Conditioning System, Free Piston, by R. J. MCCRORY, R. W. KING, and J. H. MCNINCH 377

Compressors, Estimating Water Content of Certain Dried but Uncharged Hermetic Refrigerating, by H. M. ELSEY, J. B. KELLEY, and R. B. SHARPE.... 651
Compressors Used on Household Refrigerators, Some Methods for Investigating Noise from, by R. C. BINDER.... 471
Computer for Fan Testing, Automatic, by C. H. POUNTNEY, JR. and D. W. SKIPWORTH.... 393
Condensation between the Panes of Double Windows, by A. GRANT WILSON and E. S. NOWAK.... 551
Discussion.... 570
Conductivity Measurement of Dry and Moist Materials, Thermal, by W. WOODSIDE.... 291
Conferences (ASHRAE)
Cryogenics (V. J. JOHNSON, Chairman).... 343
Domestic Refrigerator Engineering (F. A. NOLL, Chairman).... 342
Industrial Ventilation (G. B. PRIESTER, Moderator).... 342
Consolidated By-Laws.... 18
Consolidation Documents.... 13
Consolidation, Story of.... 1
Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems, by J. W. ZAHRADNIK.... 589
Discussion.... 596
Cooling, Evaluation of Three Room Air Distribution Systems for Summer, by J. J. REINMANN, ALFRED KOESTEL, and G. L. TUVE.... 717
Cooling for Common Storages of Fruits and Vegetables, Evaporative, by R. S. ASH 571
Corrosion and Water Treatment Bulletin.... 76
Council (ASHAE), Report of.... 60
CRAMER, R. D., and L. W. NEUBAUER, Solar Radiant Gains through Directional Glass Exposure.... 499
Cryogenics Bulletin.... 343
Cylinder Surface in Humid Air Cross Flow, Frost Formation and Heat Transfer on a:
Part I—Experimental Study.... 213
Part II—Theoretical Study and Conclusions.... 229

D

DANIELSON, W. A., Discussion on Evaporative Cooling for Common Storages of Fruits and Vegetables.... 587
DANIELSON, W. R., A. F. MARTZ, JR., and J. L. MARTIN, Acoustical Testing of Mufflers for Halogenated Hydrocarbon and Oil Refrigeration Systems.... 479
Determination of the Effectiveness of Window Shading Materials on the Reduction of Solar Radiation Heat Gain, by R. C. JORDAN and J. L. THRELKELD.. 683
Discussion.... 695
DIVERS, R. T., Discussion on Estimating Water Content of Certain Dried but Uncharged Hermetic Refrigerating Compressors.... 660
Discussion on Repeated Scrape Abrasion Testing of Enameled Wires in Gaseous and Liquid Refrigerants.... 682
Documents, Consolidation.... 13
DOEG, R. W., Discussion on Investigation and Control of Refrigerator Noise.... 468
DORSEY, W. G., JR., An Analysis of Turbine Driven Centrifugal Refrigeration Systems.... 349
DORSEY, W. G., JR., Discussion on Effects of Refrigerant Properties on Centrifugal Compressor Impeller Dimensions and Stage Performance.... 374
Downward Heat Transfer through a Joist Space, by P. M. CHUNG and C. E. LUND 245
Discussion.... 258
DRAKE, W. B., HARRY BUCHBERG, and D. LEBELL, Load Calculations Using Pretabulated Admittance Functions.... 515
Transfer Admittance Functions for Typical Composite Wall Sections.... 523
Dried but Uncharged Hermetic Refrigerating Compressors, Estimating Water Content of Certain, by H. M. ELSEY, J. B. KELLEY, and R. B. SHARPE.... 651
Dry and Moist Materials, Thermal Conductivity Measurement of, by W. WOODSIDE.... 291

E

Effect of Fuel Composition on Oil Burner Noise, by W. A. BEACH, R. W. SAGE, and H. F. SCHROEDER ... 627
Discussion ... 632
Effect of Pressure and Temperature on the Viscosity of Refrigerants in the Vapor Phase, by C. Z. KAMIEN and O. W. WITZELL ... 663
Effects of Refrigerant Properties on Centrifugal Compressor Impeller Dimensions and Stage Performance, by F. J. WIESNER, JR. and H. E. CASWELL ... 355
Discussion ... 374
EISEMAN, B. J., JR., Comparison of Fluoroalkane Absorption Refrigerants ... 635
EISEMAN, B. J., JR., Discussion on Repeated Scrape Abrasion Testing of Enameled Wires in Gaseous and Liquid Refrigerants ... 681
Electric Space Heating, Annual Energy Use for, by R. L. BOYD ... 259
ELSEY, H. M., J. B. KELLEY, and R. B. SHARPE, Estimating Water Content of Certain Dried but Uncharged Hermetic Refrigerating Compressors ... 651
Enameled Wires in Gaseous and Liquid Refrigerants, Repeated Scrape Abrasion Testing of, by F. F. TRUNZO, G. J. BICH, and G. W. HEWITT ... 675
Environment Reactions in the 80 to 105 F Zone, by B. H. JENNINGS and BARUCH GIVONI ... 115
Discussion ... 129
Environments, Metabolic Patterns in Stressful, by NELLO PACE ... 101
Estimating Water Content of Certain Dried but Uncharged Hermetic Refrigerating Compressors, by H. M. ELSEY, J. B. KELLEY, and R. B. SHARPE ... 651
Discussion ... 660
Evaluation of Filters for Removing Irritants from Polluted Air, by N. A. RICHARDSON and W. C. MIDDLETON ... 401
Discussion ... 415
Evaluation of Three Room Air Distribution Systems for Summer Cooling, by J. J. REINMANN, ALFRED KOESTEL, and G. L. TUVE ... 717
Discussion ... 732
Evaporative Cooling for Common Storages of Fruits and Vegetables, by R. S. ASH ... 571
Discussion ... 587
EVERETTS, JOHN, JR., Discussion on Annual Energy Use for Electric Space Heating 266
Experimental Study of Grille Noise Characteristics, by B. H. MARVET ... 613
Discussion ... 625

F

Fan Testing, Automatic Computer for, by C. H. POUNTNEY, JR. and D. W. SKIPWORTH ... 393
Filters for Removing Irritants from Polluted Air, Evaluation of, by N. A. RICHARDSON and W. C. MIDDLETON ... 401
FLACK, PETER, Discussion on Environment Reactions in the 80 to 105 F Zone ... 132
FLEISHER, WALTER L., In Memoriam (1880–1959) ... 747
Fluoroalkane Absorption Refrigerants, Comparison of, by B. J. EISEMAN, JR. ... 635
FONTAINE, W. E., Discussion on Investigation and Control of Refrigerator Noise ... 468
Free Piston Compressor in an Air-Conditioning System, by R. J. MCCRORY, R. W. KING, and J. H. MCNINCH ... 377
Discussion ... 391
Frost Formation and Heat Transfer on a Cylinder Surface in Humid Air Cross Flow:
Part I—Experimental Study ... 213
Part II—Theoretical Study and Conclusions ... 229
Fruits and Vegetables, Evaporative Cooling for Common Storages of, by R. S. ASH 571
Fuel Composition on Oil Burner Noise, Effect of, by W. A. BEACH, R. W. SAGE, and H. F. SCHROEDER ... 627
Fundamental Analysis of Chimney Performance, by W. G. COLBORNE and W. C. MOFFATT ... 417
Discussion ... 439

G

Gaseous and Liquid Refrigerants, Repeated Scrape Abrasion Testing of Enameled Wires in, by F. F. TRUNZO, G. J. BICH, and G. W. HEWITT 675
Gas-Fired Residential Heating Equipment, Pulsations in Single-Port, by C. F. SPEICH and A. A. PUTNAM 153
Gas-Fired Residential Heating Equipment, Suppression of Oscillations in, by C. F. SPEICH and A. A. PUTNAM 489
GEBHART, BENJAMIN, A New Method for Calculating Radiant Exchanges 321
GIVONI, BARUCH, and B. H. JENNINGS, Environment Reactions in the 80 to 105 F Zone 115
Glass Exposure, Solar Radiant Gains through Directional, by R. D. CRAMER and L. W. NEUBAUER 499
Glass with Roller Shades, Heat Flow through, by NECATI OZISIK and L. F. SCHUTRUM 695
GOETZ, M. B., Discussion on Free Piston Compressor in an Air-Conditioning System 391
GRAHAM, J. B., Discussion on Automatic Computer for Fan Testing 400
Grille Noise Characteristics, Experimental Study of, by B. H. MARVET 613

H

HAINES, J. E., Discussion on Evaluation of Filters for Removing Irritants from Polluted Air 415
Heat-Flow Measuring Apparatus, Large-Scale Wall, by K. R. SOLVASON 541
Heat Flow through Glass with Roller Shades, by NECATI OZISIK and L. F. SCHUTRUM 697
Discussion 715
Heat Gain, Determination of the Effectiveness of Window Shading Materials on the Reduction of Solar Radiation, by R. C. JORDAN and J. L. THRELKELD 683
Heat Gain through Windows Shaded by Metal Awnings, by NECATI OZISIK and L. F. SCHUTRUM 311
Discussion 320
Heat Pump Performance Bulletin 76
Heat Transfer on a Cylinder Surface in Humid Air Cross Flow, Frost Formation and:
Part I—Experimental Study 213
Part II—Theoretical Study and Conclusions 229
Heat Transfer through a Joist Space, Downward, by P. M. CHUNG and C. E. LUND 245
Heating and Ventilation in Relation to Human Comfort, Research on, by THOMAS BEDFORD 83
Heating, Annual Energy Use for Electric Space, by R. L. BOYD 259
Heating by Hypocaust, by EDWIN D. THATCHER 269
Discussion 288
Heating Equipment, Pulsations in Single-Port Gas-Fired Residential, by C. F. SPEICH and A. A. PUTNAM 153
Heating Equipment, Suppression of Oscillations in Gas-Fired Residential, by C. F. SPEICH and A. A. PUTNAM 489
Heating, Supply Outlet Locations for Basement, by J. R. WRIGHT and D. R. BAHNFLETH 137
Hermetic Refrigerating Compressors, Estimating Water Content of Certain Dried but Uncharged, by H. M. ELSEY, J. B. KELLEY, and R. B. SHARPE 651
HERRINGTON, L. P., Discussion on Research on Heating and Ventilation in Relation to Human Comfort 97
HEWITT, G. W., F. F. TRUNZO, and G. J. BICH, Repeated Scrape Abrasion Testing of Enameled Wires in Gaseous and Liquid Refrigerants 675
HICK, F. K., M.D., Discussion on Metabolic Patterns in Stressful Environments 113
HIERS, C. R., Discussion on Evaluation of Filters for Removing Irritants from Polluted Air 415
Discussion on Proposed ASHRAE Standards for the Measurement of Sound from Equipment 611
HOPPER, W. F., Surface Odor Adsorption and Retention Properties of Surfaces 735

HORNE, GEORGE A., In Memoriam (1877–1959) 749
Hot-Wire Anemometer, Omni-Directional Low-Speed, by J. F. KEMP 205
HUEBNER, W. O., Discussion on Heating by Hypocaust 288
Discussion on Omni-Directional Low-Speed Hot-Wire Anemometer 212
Humid Air Cross Flow, Frost Formation and Heat Transfer on a Cylinder Surface in:
Part I—Experimental Study 213
Part II—Theoretical Study and Conclusions 229
HURTGEN, J. P., Discussion on Estimating Water Content of Certain Dried but Uncharged Hermetic Refrigerating Compressors 661
Hydrocarbon and Oil Refrigeration Systems, Acoustical Testing of Mufflers for Halogenated, by A. F. MARTZ, JR., J. L. MARTIN, and W. R. DANIELSON .. 479
Hydronics Bulletin 75
Hypocaust, Heating by, by EDWIN D. THATCHER 269

I

In Memoriam 745
Industrial Ventilation Bulletin 342
Inspectors of Election, Report of 74
Investigation and Control of Refrigerator Noise, by E. A. BAILLIF and J. P. LAUGHLIN 443
Discussion 468
Irritants from Polluted Air, Evaluation of Filters for Removing, by N. A. RICHARDSON and W. C. MIDDLETON 401

J

JENNINGS, B. H., and BARUCH GIVONI, Environment Reactions in the 80 to 105 F Zone 115
JENSON, E. H., Discussion on Proposed ASHRAE Standards for the Measurement of Sound from Equipment 611
JORDAN, R. C., and J. L. THRELKELD, Determination of the Effectiveness of Window Shading Materials on the Reduction of Solar Radiation Heat Gain . 683

K

KAMIEN, C. Z., and O. W. WITZELL, Effect of Pressure and Temperature on the Viscosity of Refrigerants in the Vapor Phase 663
KELLEY, J. B., Discussion on Repeated Scrape Abrasion Testing of Enamelled Wires in Gaseous and Liquid Refrigerants 681
KELLEY, J. B., R. B. SHARPE, and H. M. ELSEY, Estimating Water Content of Certain Dried but Uncharged Hermetic Refrigerating Compressors 651
KEMP, J. F., Omni-Directional Low-Speed Hot-Wire Anemometer 205
KENNEDY, W. W., Discussion on Experimental Study of Grille Noise Characteristics 626
KERKA, W. T., Discussion on Experimental Study of Grille Noise Characteristics 625
KING, R. W., J. H. MCNINCH, and R. J. MCCRORY, Free Piston Compressor in an Air-Conditioning System 377
KINNEY, HARRY, Discussion on Experimental Study of Grille Noise Characteristics 626
KOESTEL, ALFRED, Discussion on Omni-Directional Low-Speed Hot-Wire Anemometer 212
KOESTEL, ALFRED, G. L. TUVE, and J. J. REINMANN, Evaluation of Three Room Air Distribution Systems for Summer Cooling 717

L

Large-Scale Wall Heat-Flow Measuring Apparatus, by K. R. SOLVASON 541
Discussion 550
LATTER, BRUCE, Discussion on Investigation and Control of Refrigerator Noise. 469
LAUGHLIN, J. P., and E. A. BAILLIF, Investigation and Control of Refrigerator Noise 443

LAUGHLIN, J. P., Discussion on Proposed ASHRAE Standards for the Measurement of Sound from Equipment 610
LAWRIE, W. E., and G. J. SANDERS, Low Frequency Combustion Noise in Oil-Burning Equipment 181
LEBELL, D., W. B. DRAKE, and HARRY BUCHBERG, Load Calculations Using Pretabulated Admittance Functions 515
Transfer Admittance Functions for Typical Composite Wall Sections 523
LEE, B. H., Discussion on An Analysis of Turbine Driven Centrifugal Refrigeration Systems 353
LEE, J. C., Discussion on An Analysis of Turbine Driven Centrifugal Refrigeration Systems 353
Discussion on Comparison of Fluoroalkane Absorption Refrigerants 650
Discussion on Effects of Refrigerant Properties on Centrifugal Compressor Impeller Dimensions and Stage Performance 375
LEOPOLD, C. S., Discussion on Environment Reactions in the 80 to 105 F Zone .. 129
Discussion on Research on Heating and Ventilation in Relation to Human Comfort 99
LIEPIN, LEE, Discussion on Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems 596
LINSENMEYER, F. J., Discussion on Heating by Hypocaust 288
LIPSTEIN, NORMAN J., Discussion on Annual Energy Use for Electric Space Heating 267
Liquid Refrigerants, Repeated Scrape Abrasion Testing of Enameled Wires in Gaseous and, by F. F. TRUNZO, G. J. BICH, and G. W. HEWITT 675
Load Calculations Using Pretabulated Admittance Functions, by W. B. DRAKE, HARRY BUCHBERG, and D. LEBELL 515
Discussion 540
LOCKLIN, D. W., C. F. SPEICH and A. A. PUTNAM, Joint Discussion on Low Frequency Combustion Noise in Oil-Burning Equipment 200
LOCKLIN, D. W., Discussion on A New Method of Chimney Design and Performance Evaluation 440
Discussion on Effect of Fuel Composition on Oil Burner Noise 632
Low Frequency Combustion Noise in Oil-Burning Equipment, by G. J. SANDERS and W. E. LAWRIE 181
Discussion 199
Low-Speed Hot-Wire Anemometer, Omni-Directional, by J. F. KEMP 205
LUND, C. E., and P. M. CHUNG, Downward Heat Transfer through a Joist Space. 245

M

MACORMACK, J. G., Discussion on Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems 597
Discussion on Large-Scale Wall Heat-Flow Measuring Apparatus 550
MARTIN, J. L., W. R. DANIELSON, and A. F. MARTZ, JR., Acoustical Testing of Mufflers for Halogenated Hydrocarbon and Oil Refrigeration Systems 479
MARTZ, A. F., JR., J. L. MARTIN, and W. R. DANIELSON, Acoustical Testing of Mufflers for Halogenated Hydrocarbon and Oil Refrigeration Systems 479
MARVET, B. H., Experimental Study of Grille Noise Characteristics 613
MCCRORY, R. J., R. W. KING, and J. H. MCNINCH, Free Piston Compressor in an Air-Conditioning System 377
MCKINLEY, R. W., Discussion on Condensation between the Panes of Double Windows 570
Discussion on Determination of the Effectiveness of Window Shading Materials on the Reduction of Solar Radiation Heat Gain 696
Discussion on Evaluation of Three Room Air Distribution Systems for Summer Cooling 732
Discussion on Heat Flow through Glass with Roller Shades 715
MCNINCH, J. H., R. J. MCCRORY, and R. W. KING, Free Piston Compressor in an Air-Conditioning System 377
MEDON, F. E., Discussion on Free Piston Compressor in an Air-Conditioning System 391
Meeting Arrangements Committee, ASHRAE Annual Meeting, 1959 (Lake Placid, New York) 348

Metabolic Patterns in Stressful Environments, by NELLO PACE 101
Discussion 113
Metal Awnings, Heat Gain through Windows Shaded by, by NECATI OZISIK and L. F. SCHUTRUM 311
MIDDLETON, W. C., and N. A. RICHARDSON, Evaluation of Filters for Removing Irritants from Polluted Air 410
MILLER, V. L., Discussion on Heat Gain through Windows Shaded by Metal Awnings 320
MIN, T. C., Discussion on A New Method for Calculating Radiant Exchanges .. 330
Discussion on Evaluation of Three Room Air Distribution Systems for Summer Cooling 733
MITTENDORFF, E. M., Discussion on Low Frequency Combustion Noise in Oil-Burning Equipment 199
MOFFATT, W. C., and W. G. COLBORNE, A New Method of Chimney Design and Performance Evaluation 431
Fundamental Analysis of Chimney Performance 417
Moist Materials, Thermal Conductivity Measurement of Dry and, by W. WOODSIDE 291
Morning After for Education, The, by DR. ERIC A. WALKER 333
Mufflers for Halogenated Hydrocarbon and Oil Refrigeration Systems, Acoustical Testing of, by A. F. MARTZ, JR., J. L. MARTIN, and W. R. DANIELSON 479
MUNGER, G. R., Discussion on Determination of the Effectiveness of Window Shading Materials on the Reduction of Solar Radiation Heat Gain 696
Discussion on Heat Flow through Glass with Roller Shades 715
Discussion on Load Calculations Using Pretabulated Admittance Functions and Transfer Admittance Functions for Typical Composite Wall Sections .. 540
MYLER, W. M., Discussion on Low Frequency Combustion Noise in Oil-Burning Equipment 199

N

NEUBAUER, L. W., and R. D. CRAMER, Solar Radiant Gains through Directional Glass Exposure 499
NEVINS, R. C., Discussion on Environment Reactions in the 80 to 105 F Zone .. 129
New Method for Calculating Radiant Exchanges, A, by BENJAMIN GEBHART 321
Discussion 329
New Method of Chimney Design and Performance Evaluation, A, by W. C. MOFFAT and W. G. COLBORNE 431
Discussion 439
Noise Characteristics, Experimental Study of Grille, by B. H. MARVET 613
Noise, Effect of Fuel Composition on Oil Burner, by W. A. BEACH, R. W. SAGE, and H. F. SCHROEDER 627
Noise from Compressors Used on Household Refrigerators, Some Methods for Investigating, by R. C. BINDER 471
Noise in Oil-Burning Equipment, Low Frequency Combustion, by G. J. SANDERS and W. E. LAWRIE 181
Noise, Investigation and Control of Refrigerator, by E. A. BAILLIF and J. P. LAUGHLIN 443
Notice of Special Meeting to all Members Entitled to Vote 10
NOTTAGE, H. B., Discussion on Supply Outlet Locations for Basement Heating. 151
NOWAK, E. S., and A. GRANT WILSON, Condensation between the Panes of Double Windows 551

O

Odor Adsorption and Retention Properties of Surfaces, Surface, by W. F. HOPPER 735
Officers and Board of Directors (January 1959 to June 1959) 31
Officers and Board of Directors (June 1959 to January 1961) 32
Officers and Board of Directors (June 1961 to June 1962) 33
Oil Burner Noise, Effect of Fuel Composition on, by W. A. BEACH, R. W. SAGE, and H. F. SCHROEDER 627
Oil-Burning Equipment, Low Frequency Combustion Noise in, by G. J. SANDERS and W. E. LAWRIE 181

Oil Refrigeration Systems, Acoustical Testing of Mufflers for Halogenated Hydrocarbon and, by A. F. MARTZ, JR., J. L. MARTIN, and W. R. DANIELSON 479
OLSON, EUGENE O., Discussion on Low Frequency Combustion Noise in Oil-Burning Equipment 202
Omni-Directional Low-Speed Hot-Wire Anemometer, by J. F. KEMP 205
Discussion 211
Order of Court for Consolidation 39
Oscillations in Gas-Fired Residential Heating Equipment, Suppression of, by C. F. SPEICH and A. A. PUTNAM 489
OURUSOFF, LEONID, Discussion on Comparison of Fluoroalkane Absorption Refrigerants 650
Outlet Locations for Basement Heating, Supply, by J. R. WRIGHT and D. R. BAHNFLETH 137
OZISIK, NECATI, and L. F. SCHUTRUM, Heat Flow through Glass with Roller Shades 695
Heat Gain through Windows Shaded by Metal Awnings 311

P

PACE, NELLO, Metabolic Patterns in Stressful Environments 101
Panes of Double Windows, Condensation between the, by A. GRANT WILSON and E. S. NOWAK 551
PENTZER, W. T., Discussion on Evaporative Cooling for Common Storages of Fruits and Vegetables 587
Petition for Consolidation 33
Piston Compressor in an Air-Conditioning System, Free, by R. J. MCCRORY, R. W. KING, and J. H. MCNINCH 377
PLOEGER, C. E., Discussion on Acoustical Testing of Mufflers for Halogenated Hydrocarbon and Oil Refrigeration Systems 487
Polluted Air, Evaluation of Filters for Removing Irritants from, by N. A. RICHARDSON and W. C. MIDDLETON 401
POPYK, W. W., Discussion on Low Frequency Combustion Noise in Oil-Burning Equipment 200
Discussion on Pulsations in Single-Port Gas-Fired Residential Heating Equipment 179
POTTER, C. D., Discussion on Effect of Fuel Composition on Oil Burner Noise 633
Discussion on Solar Radiant Gains through Directional Glass Exposure 513
POUNTNEY, C. H., JR., and D. W. SKIPWORTH, Automatic Computer for Fan Testing 393
President, Report of (Cecil Boling, ASHRAE) 341
President, Report of (E. R. Queer, ASHAE) 49
Pressure and Temperature on the Viscosity of Refrigerants in the Vapor Phase, Effect of, by C. Z. KAMIEN and O. W. WITZELL 663
Pretabulated Admittance Functions, Load Calculations Using, by W. B. DRAKE, HARRY BUCHBERG, and D. LEBELL 515
PRIESTER, G. B., Discussion on Evaluation of Filters for Removing Irritants from Polluted Air 415
Probe for Thermal Conductivity Measurement of Dry and Moist Materials, by W. WOODSIDE 291
Program, ASHRAE Annual Meeting, 1959, (Lake Placid, New York) 343
65th Annual Meeting ASHAE 78
Proposed ASHRAE Standards for the Measurement of Sound from Equipment, by C. M. ASHLEY 601
Discussion 610
Provisions of Certificate of Incorporation of ASHAE 36
Provisions of Certificate of Incorporation of ASRE 38
PRUCHA, R. V., Discussion on Load Calculations Using Pretabulated Admittance Functions and Transfer Admittance Functions for Typical Composite Wall Sections 540
Pulsations in Single-Port Gas-Fired Residential Heating Equipment, by C. F. SPEICH and A. A. PUTNAM 153
Discussion 179

PUTNAM, A. A., and C. F. SPEICH, Pulsations in Single-Port Gas-Fired Residential Heating Equipment 153
Suppression of Oscillations in Gas-Fired Residential Heating Equipment 489
PUTNAM, A. A., D. W. LOCKLIN and C. F. SPEICH, Joint Discussion on Low Frequency Combustion Noise in Oil-Burning Equipment 200

Q

QUEER, E. R., Report of President (ASHAE) 49

R

Radiant Exchanges, A New Method for Calculating, by BENJAMIN GEBHART 321
Radiant Gains through Directional Glass Exposure, Solar, by R. D. CRAMER and L. W. NEUBAUER 499
Radiation Heat Gain, Determination of the Effectiveness of Window Shading Materials on the Reduction of Solar, by R. C. JORDAN and J. L. THRELKELD 683
Reactions in the 80 to 105 F Zone, Envinroment, by B. H. JENNINGS and BARUCH GIVONI 115
Refrigerant Properties on Centrifugal Compressor Impeller Dimensions and Stage Performance, Effects of, by F. J. WIESNER, JR. and H. E. CASWELL .. 355
Refrigerants, Comparison of Fluoroalkane Absorption, by B. J. EISEMAN, JR. .. 635
Refrigerants in the Vapor Phase, Effect of Pressure and Temperature on the Viscosity of, by C. Z. KAMIEN and O. W. WITZELL 663
Refrigerants, Repeated Scrape Abrasion Testing of Enameled Wires in Gaseous and Liquid, by F. F. TRUNZO, G. J. BICH, and G. W. HEWITT 675
Refrigerating Compressors, Estimating Water Content of Certain Dried but Uncharged Hermetic, by H. M. ELSEY, J. B. KELLEY, and R. B. SHARPE 651
Refrigeration Structures and Systems, Controlled Atmosphere Apple Storage Process and Its Requirements on, by J. W. ZAHRADNIK 589
Refrigeration Systems, Acoustical Testing of Mufflers for Halogenated Hydrocarbon and Oil, by A. F. MARTZ, JR., J. L. MARTIN, and W. R. DANIELSON .. 479
Refrigeration Systems, An Analysis of Turbine Driven Centrifugal, by W. G. DORSEY, JR. 349
Refrigerator Noise, Investigation and Control of, by E. A. BAILLIF and J. P. LAUGHLIN 443
Refrigerators, Some Methods for Investigating Noise from Compressors Used on Household, by R. C. BINDER 471
REICH, H. H., Discussion on Supply Outlet Locations for Basement Heating 151
REINMANN, J. J., ALFRED KOESTEL, and G. L. TUVE, Evaluation of Three Room Air Distribution Systems for Summer Cooling 717
Repeated Scrape Abrasion Testing of Enameled Wires in Gaseous and Liquid Refrigerants, by F. F. TRUNZO, G. J. BICH, and G. W. HEWITT 675
Discussion 681
Report of Council (ASHAE) 60
Report of President (C. E. Boling, ASHRAE) 341
Report of President (E. R. Queer, ASHAE) 49
Research on Heating and Ventilation in Relation to Human Comfort, by THOMAS BEDFORD 83
Discussion 97
Residential Heating Equipment, Pulsations in Single-Port Gas-Fired, by C. F. SPEICH and A. A. PUTNAM 153
Residential Heating Equipment, Suppression of Oscillations in Gas-Fired, by C. F. SPEICH and A. A. PUTNAM 489
Resolutions Committee, Report of (65th Annual Meeting, ASHAE) 76
RICHARDSON, N. A., and W. C. MIDDLETON, Evaluation of Filters for Removing Irritants from Polluted Air 401
ROBERTSON, E. J., Discussion on Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems 596
Room Air Distribution Systems for Summer Cooling, Evaluation of Three, by J. J. REINMANN, ALFRED KOESTEL, and G. L. TUVE 717
ROUSHEY, R. E., Discussion on An Analysis of Turbine Driven Centrifugal Refrigeration Systems 353
ROZEN, G., Discussion on Environment Reactions in the 80 to 105 F Zone 131

S

SAGE, R. W., Discussion on Low Frequency Combustion Noise in Oil-Burning Equipment ... 202
SAGE, R. W., H. F. SCHROEDER, and W. A. BEACH, Effect of Fuel Composition on Oil Burner Noise ... 627
SANDERS, G. J., and W. E. LAWRIE, Low Frequency Combustion Noise in Oil-Burning Equipment ... 181
SAWYER, G. N., Discussion on Proposed ASHRAE Standards for the Measurement of Sound from Equipment ... 610
SCHROEDER, H. F., W. A. BEACH, and R. W. SAGE, Effect of Fuel Composition on Oil Burner Noise ... 627
SCHUTRUM, L. F., and NECATI OZISIK, Heat Flow through Glass with Roller Shades ... 695
SCHUTRUM, L. F., Discussion on Determination of the Effectiveness of Window Shading Materials on the Reduction of Solar Radiation Heat Gain ... 695
Heat Gain through Windows Shaded by Metal Awnings ... 311
Discussion on Solar Radiant Gains through Directional Glass Exposure ... 513
Shades, Heat Flow through Glass with Roller, by NECATI OZISIK and L. F. SCHUTRUM ... 695
SHARPE, R. B., H. M. ELSEY, and J. B. KELLEY, Estimating Water Content of Certain Dried but Uncharged Hermetic Refrigerating Compressors ... 651
Single-Port Gas-Fired Residential Heating Equipment, Pulsations in, by C. F. SPEICH and A. A. PUTNAM ... 153
Sixty-fifth Annual Meeting, 1959, ASHAE (Philadelphia, Pa.) ... 47
SKIPWORTH, D. W., and C. H. POUNTNEY, JR., Automatic Computer for Fan Testing ... 393
SMITH, W. T., Discussion on Evaporative Cooling for Common Storages of Fruits and Vegetables ... 588
Solar Radiant Gains through Directional Glass Exposure, by R. D. CRAMER and L. W. NEUBAUER ... 499
Discussion ... 513
Solar Radiation Heat Gain, Determination of the Effectiveness of Window Shading Materials on the Reduction of, by R. C. JORDAN and J. L. THRELKELD ... 683
SOLING, S. P., Discussion on Effects of Refrigerant Properties on Centrifugal Compressor Impeller Dimensions and Stage Performance ... 376
SOLVASON, K. R., Discussion on A New Method for Calculating Radiant Exchanges ... 329
Discussion on Supply Outlet Locations for Basement Heating ... 152
SOLVASON, K. R., Large-Scale Wall Heat-Flow Measuring Apparatus ... 541
Some Methods for Investigating Noise from Compressors Used on Household Refrigerators, by R. C. BINDER ... 471
Sound from Equipment, Proposed ASHRAE Standards for the Measurement of, by C. M. ASHLEY ... 601
SPAUSCHUS, H. O., Discussion on Comparison of Fluoroalkane Absorption Refrigerants ... 649
Special Meeting December 1, 1958 (Chicago, Ill.) ... 9
SPEICH, C. F., A. A. PUTNAM and D. W. LOCKLIN, Joint Discussion on Low Frequency Combustion Noise in Oil-Burning Equipment ... 200
SPEICH, C. F., and A. A. PUTNAM, Pulsations in Single-Port Gas-Fired Residential Heating Equipment ... 153
Suppression of Oscillations in Gas-Fired Residential Heating Equipment ... 489
STACHELEK, S. J., Discussion on Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems ... 597
Standards for the Measurement of Sound from Equipment, Proposed ASHRAE, by C. M. ASHLEY ... 601
Storage Process and Its Requirements on Refrigeration Structures and Systems, Controlled Atmosphere Apple, by J. W. ZAHRADNIK ... 589
Storages of Fruits and Vegetables, Evaporative Cooling for Common, by R. S. ASH 571
Story of Consolidation ... 1
Summer Cooling, Evaluation of Three Room Air Distribution Systems for, by J. J. REINMANN, ALFRED KOESTEL, and G. L. TUVE ... 717

SUMMERHAYS, R. L., Discussion on Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems 597
Discussion on Evaporative Cooling for Common Storages of Fruits and Vegetables 587
Supply Outlet Locations for Basement Heating, by J. R. WRIGHT and D. R. BAHNFLETH 137
Discussion 151
Suppression of Oscillations in Gas-Fired Residential Heating Equipment, by C. F. SPEICH and A. A. PUTNAM 489
Surface Odor Adsorption and Retention Properties of Surfaces, by W. F. HOPPER. 735
SWANTON, J. R., JR., Discussion on Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems 596
Symposiums on (ASHAE)
Corrosion and Water Treatment (W. G. HOLE, Moderator) 76
Heat Pump Performance (G. B. PRIESTER, Moderator) 76
Hydronics (P. N. VINTHER, Moderator) 75

T

TAC Activities (ASHAE) 65
TARLETON, F. L., Discussion on Estimating Water Content of Certain Dried but Uncharged Hermetic Refrigerating Compressors 661
Temperature on the Viscosity of Refrigerants in the Vapor Phase, Effect of Pressure and, by C. Z. KAMIEN and O. W. WITZELL 663
THATCHER, EDWIN D., Heating by Hypocaust 269
Thermal Conductivity Measurement of Dry and Moist Materials, by W. WOODSIDE 291
THRELKELD, J. L., and R. C. JORDAN, Determination of the Effectiveness of Window Shading Materials on the Reduction of Solar Radiation Heat Gain. 683
TOKACH, M. F., Discussion on Free Piston Compressor in an Air-Conditioning System 391
Transfer Admittance Functions for Typical Composite Wall Sections, by W. B. DRAKE, HARRY BUCHBERG, and D. LEBELL 523
Discussion 540
Treasurer's Report (J. H. FOX) 50
TRUNZO, F. F., G. J. BICH, and G. W. HEWITT, Repeated Scrape Abrasion Testing of Enameled Wires in Gaseous and Liquid Refrigerants 675
Turbine Driven Centrifugal Refrigeration Systems, An Analysis of, by W. G. DORSEY, JR. 349
TUVE, G. L., J. J. REINMANN, and ALFRED KOESTEL, Evaluation of Three Room Air Distribution Systems for Summer Cooling 717

V

Vapor Phase, Effect of Pressure and Temperature on the Viscosity of Refrigerants in the, by C. Z. KAMIEN and O. W. WITZELL 663
Vegetables, Evaporative Cooling for Common Storages of Fruits and, by R. S. ASH 571
Ventilation in Relation to Human Comfort, Research on Heating and, by THOMAS BEDFORD 83
VILD, D. J., Discussion on Heat Flow through Glass with Roller Shades 715

W

WALKER, DR. ERIC A., The Morning After for Education 333
WALKER, W. O., Discussion on Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems 596
Wall Heat-Flow Measuring Apparatus, Large-Scale, by K. R. SOLVASON 541
Wall Sections, Transfer Admittance Functions for Typical Composite, by W. B. DRAKE, HARRY BUCHBERG, and D. LEBELL 523
Water Content of Certain Dried but Uncharged Hermetic Refrigerating Compressors, Estimating, by H. M. ELSEY, J. B. KELLEY, and R. B. SHARPE 651

WIESNER, F. J., JR., and H. E. CASWELL, Effects of Refrigerant Properties on Centrifugal Compressor Impeller Dimensions and Stage Performance 355
WILE, D. D., Discussion on Investigation and Control of Refrigerator Noise 468
WILSON, A. G., Discussion on A New Method of Chimney Design and Performance Evaluation 440
Discussion on Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems 597
WILSON, A. GRANT, and E. S. NOWAK, Condensation between the Panes of Double Windows 551
Window Shading Materials on the Reduction of Solar Radiation Heat Gain, Determination of the Effectiveness of, by R. C. JORDAN and J. L. THRELKELD 683
Windows, Condensation between the Panes of Double, by A. GRANT WILSON and E. S. NOWAK 551
Windows Shaded by Metal Awnings, Heat Gain through, by NECATI OZISIK and L. F. SCHUTRUM 311
WITZELL, O. W., and C. Z. KAMIEN, Effect of Pressure and Temperature on the Viscosity of Refrigerants in the Vapor Phase 663
WOODSIDE, W., Thermal Conductivity Measurement of Dry and Moist Materials 291
WRIGHT, J. R., and D. R. BAHNFLETH, Supply Outlet Locations for Basement Heating 137

Z

ZAHRADNIK, J. W., Controlled Atmosphere Apple Storage Process and Its Requirements on Refrigeration Structures and Systems 589
Discussion on Solar Radiant Gains through Directional Glass Exposure .. 513